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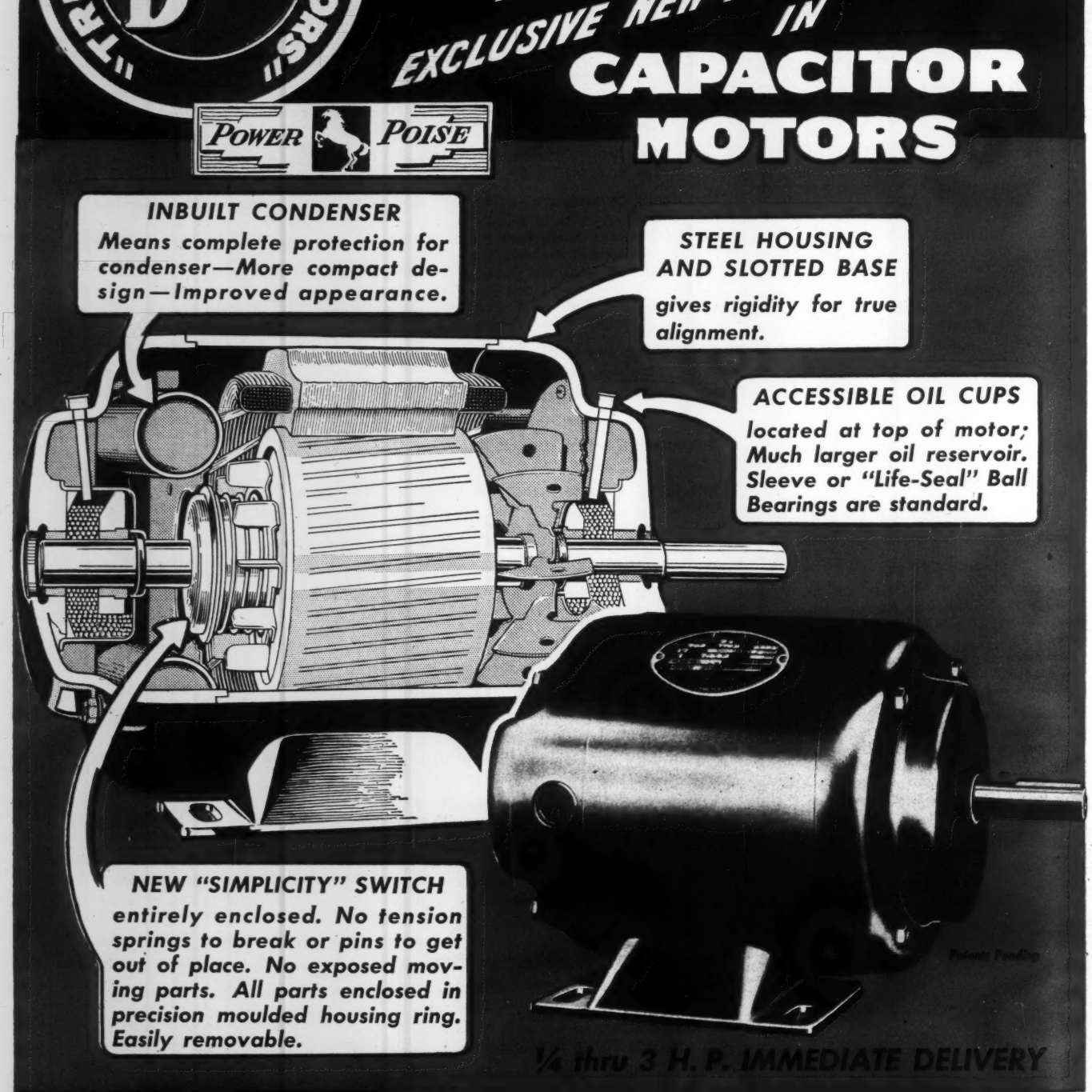

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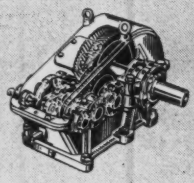

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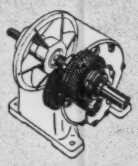

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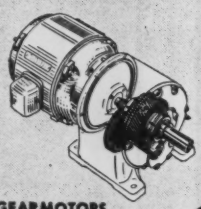

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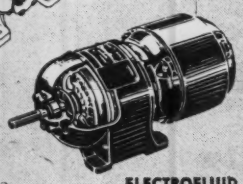

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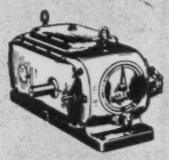

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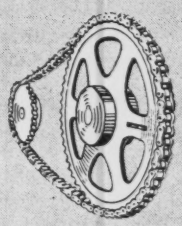

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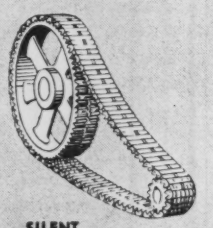

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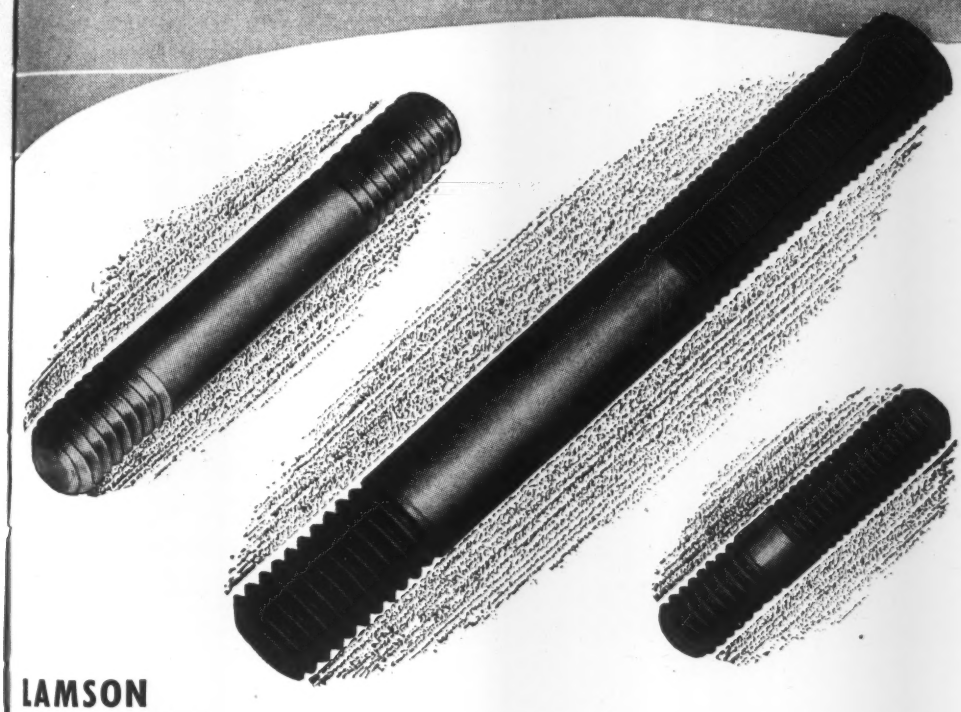

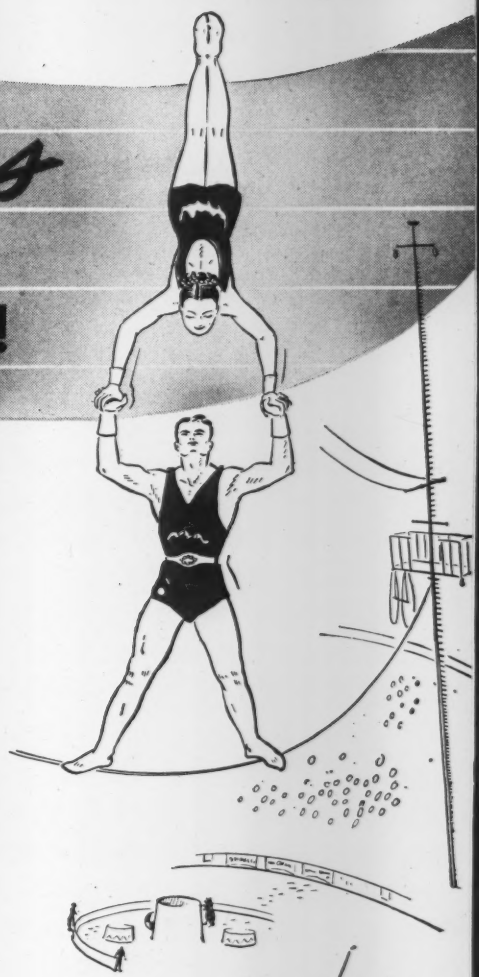

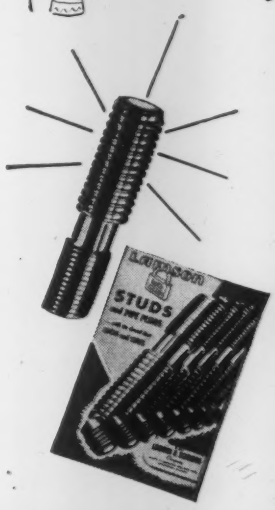

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Topics

ANTENNAE SYSTEM within the structure of the airplane itself is the result of a special modification program on an Air Force C-54 four-engine transport plane at Goodyear Aircraft Corp. To prevent the aluminum covering of the aircraft from shielding the radio waves from the antennae, sections of the surfaces involved are replaced by laminated, resin impregnated Fiberglas.

ATOMIC ENERGY fundamentals are simple, says Dr. Hogness, University of Chicago. A few terms keep many persons from understanding developments in this field. He says, if your livelihood depends upon industry, it behooves you to learn something about atomic terms.

AUTOMATIC MACHINE turns out complete two-tube radio sets, three a minute, without the intervention of human hands. A British invention, the machine requires one operator to feed molded plastic plates into a 70-foot magazine. Finished sets pop off the conveyor belt at the other end.

MAGNESIUM ANODES for cathodic protection of storage tanks prevent or greatly lessen corrosion caused by the battery reaction set up between water and the metal tank. Magnesium's high driving voltage reverses the current flow and permits the anode to be corroded rather than the tank wall.

ORIGINALLY DEVELOPED during the war to conserve critical alloying elements, NE steels account for about one-eighth of the alloy steels produced since the war. These steels have proved to be highly useful and remain in demand even though the need for conservation of alloys has been lessened.

MODEL is capable of testing, in a single day, electronic tube designs which otherwise require three months. Conceived by Westinghouse research engineers, it consists of a thin sheet of rubber stretched on a frame. Hills and valleys in the rubber simulate electrical voltage. Time for a ball to roll from one part of the frame to another is a measure of the speed of electrons and other characteristics of the tube.

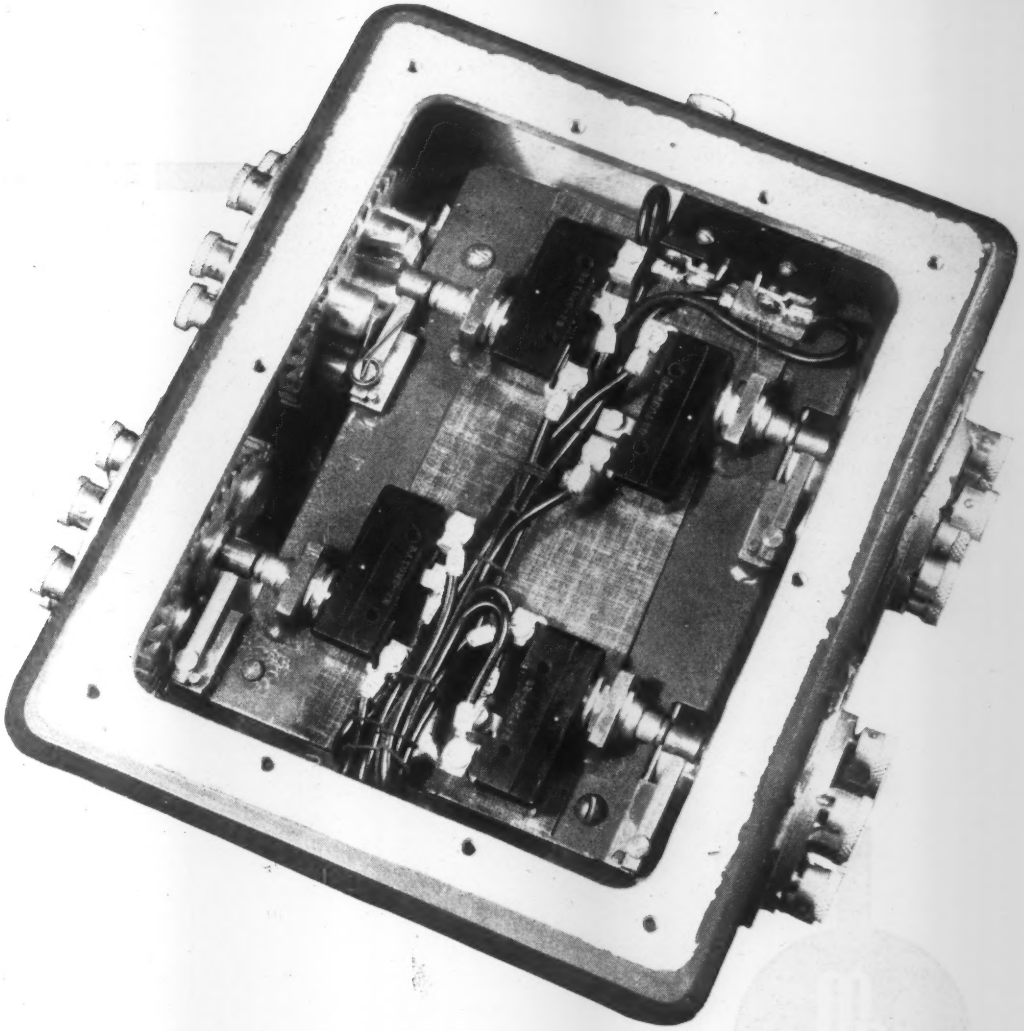
ADOPTION of a single universal pH scale to express numerically the degree of acidity or alkalinity of aqueous solutions has been recommended by the National Bureau of Standards. Four buffer materials in the form of standard samples of certified purity—acid potassium phthalate, potassium dihydrogen phosphate and disodium hydrogen phosphate, and borax—provide fixed points on the scale.

PROTONS, in the 400,000,000 electron-volt cyclotron for the University of Chicago, will be accelerated to speeds of more than 160,000 miles a second. When flung against a target at these speeds, the protons are capable of penetrating a foot into aluminum, assisting scientists in their basic studies of nuclear forces.

NEW FURNACE BRAZING techniques have been developed by General Electric Co., making possible the joining of precision, high-temperature stainless-steel parts. In the past, brazing of heat-resistant alloys containing chromium resulted in the oxidation of the chromium or, when flux was used, in deposits which were difficult to remove.

AUTOMATIC logarithmic dividing machine which will generate any logarithmic, trigonometric or linear scale up to 20 inches in length with an accuracy of 0.0002-inch was built by A. Nestler factory in Germany. The machine is 21 feet in length and is one of the finest in the world for engraving accurately and in an economical manner.

HIGH-SPEED motion camera, ten times faster than previous ones, will take more than five million pictures a second.



High-Speed Cycling

Automatic electric controls are utilized in turret lathe. Special design features reduce inertia of drive, increasing production

By R. H. Clark
Electrical Engineer
The Warner & Swasey Co.
Cleveland

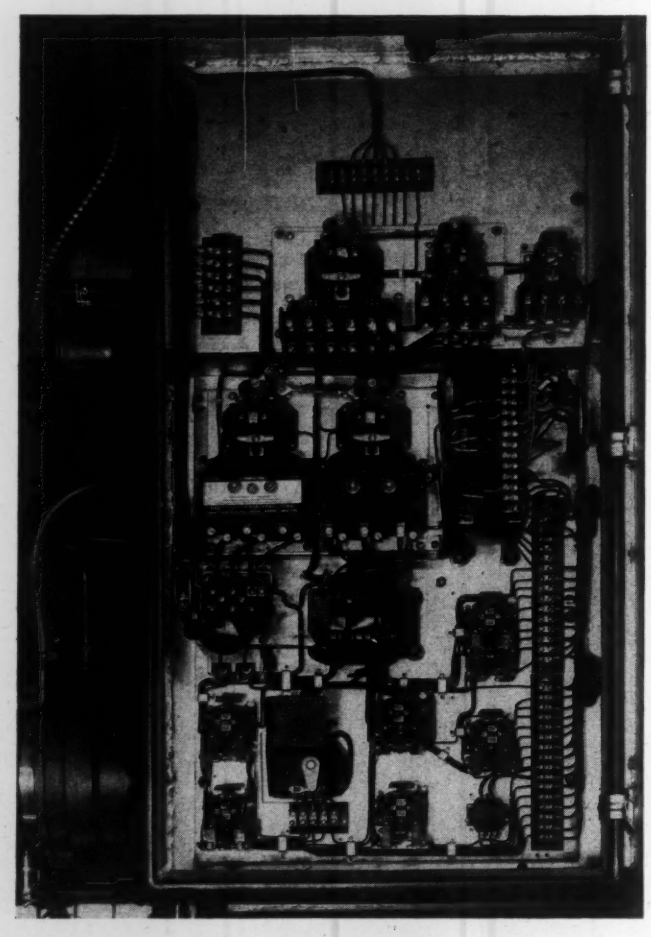
INCREASING the output of manually loaded machines requires primarily reduction in both handling time and nonproductive periods in the machine cycle. For example, an analysis of the floor-to-floor time for turret-lathe machining of many small parts—particularly of nonferrous metals—reveals that approximately 20 per cent of total time is used in actual

machining. By far the greatest amount of the remaining time is consumed in manual headstock handling, such as starting, speed changing, reversing and stopping. All of these functions have been transferred to the automatic controls shown in the head illustration and *Fig.* 1 for the small turret lathe shown in *Fig.* 2.

The automatic control panel, *Fig.* 1, receives signals from the series of limit switches located in the automatic drum control shown in the head illustration. This drum unit is mounted on the turret slide. The limit switches are operated by four drums, geared and rotatable with the rotation of the turret. Each drum contains six cam buttons, one for each turret face, preset in accordance with the desired headstock functions. As the turret is indexed to each turret face, the headstock responds with the preset functions. These automatic functions are as follows:

1. Start of spindle as tools approach work
2. Spindle speed change as turret and tools are indexed
3. Selection of spindle direction for right-hand or left-hand threading or machining
4. Reverse of spindle at end of predetermined forward turret travel for backing out taps
5. Continuation of reverse spindle direction on one or more turret faces to minimize spindle reversals (For example, following a right-hand threading operation with a left-hand reaming operation)
6. Plug-stop of spindle on indexing following the last machining operation. This function is operative for forward or back indexing, as desired, when only two or three turret faces are tooled.

Fig. 1—Automatic control panel, mounted on rear of lathe, receives signals from the drum selector. An oversize reversing starter is utilized because of the severe duty cycling



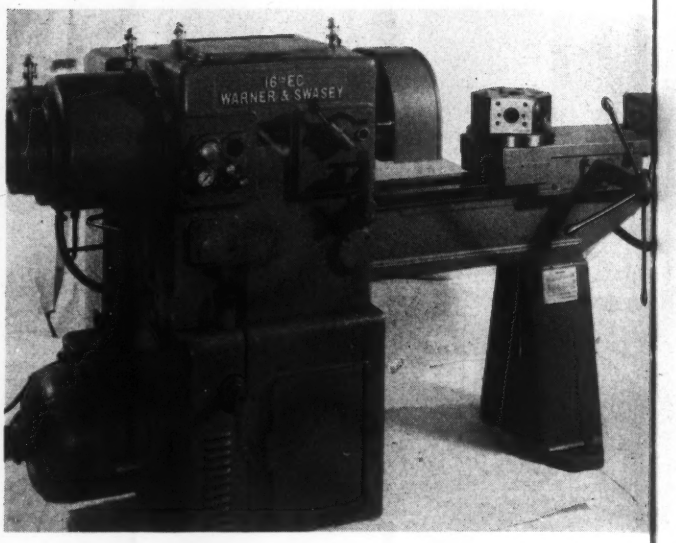


Fig. 2—Small turret lathe, designed with automatic, preset electric controls to increase production. Reversing duty in the speeded-up cycle requires special design of motor and of moving parts to reduce inertia

Auxiliary limit switches located in a watertight box attached to the back of the turret slide saddle initiate the automatic start and reverse functions, if such functions have been preset at the drum control. A cam attached to the slide actuates the start switch, and the six standard adjustable stop screws actuate the reverse switch at the desired point on the feeding stroke of the turret slide.

Precision type limit switches are used throughout to insure proper timing of the functions and accuracy of repeat of the reverse point for depth control of taps and threading tools. Reversing of the motor provides remarkable accuracy in number of revolutions to reverse after the reverse power is applied. At moderate speeds the variations, as recorded by oscillographic tests and in actual production threading, have not exceeded one-quarter revolution of the spindle. This permits using solid taps and dies, thereby reducing tooling costs and increasing accuracy.

Construction of the start-stop drum is illustrated at the upper left in the head illustration. The axis of each cam is offset 30 degrees from the limit switch plunger axis, with the corresponding turret face in machining position. All of these cams may be rotated to any one of four positions. With the pointer on the cam button in position so that the extended face of the cam holds the limit switch contact open, starting of the spindle is prevented as the turret slide is advanced toward the work. This func-

tion is used for loading plugs for seating parts securely in the chuck.

If the button pointer is turned clockwise one-quarter turn, the contact is held closed and the spindle will start as the turret slide is advanced. With successive buttons for each turret position turned to the third position, spindle rotation is maintained for as many operations as required. For stopping, the button is turned to the fourth position and, as the turret indexes from the last machining operation to the starting operation, the cam momentarily opens the contact, stopping the spindle.

Pivoted construction of the cam is used for stopping the spindle when the turret is back indexed. This condition occurs when only two or three turret faces are tooled. Time is saved by back indexing rather than forward indexing to the starting operation. With the button pointer in the second position, the side of the cam is presented to the limit switch in an axial position which would normally trip the contact and stop the spindle. However, as the cam strikes the plunger, it pivots against a spring and drops out of the way. If the turret is back indexed, the opposite face of the cam contacts the plunger and is restrained in its upright position by a stop. The contact is, therefore, opened and the spindle is stopped. A small pivoted hammer is used to prevent side thrust on the plunger of the limit switch, thereby lengthening its life to millions of operations.

The other three drums operate similarly, with the exception that solid cam faces are used because of the simpler functions. Axes of the cams are in line with the plungers, turret being in machining position.

Spindle positioning for easier loading of the chuck operates when the spindle stops. This is accomplished by automatically plugging the motor to a stop, applying a reduced voltage to the motor to rotate the spindle slowly, and stopping with an electric brake as controlled by a limit switch actuated by an adjustable cam on the spindle.

Control panel, *Fig.* 1, includes two-speed and reversing motor starters, spindle positioning and 110-

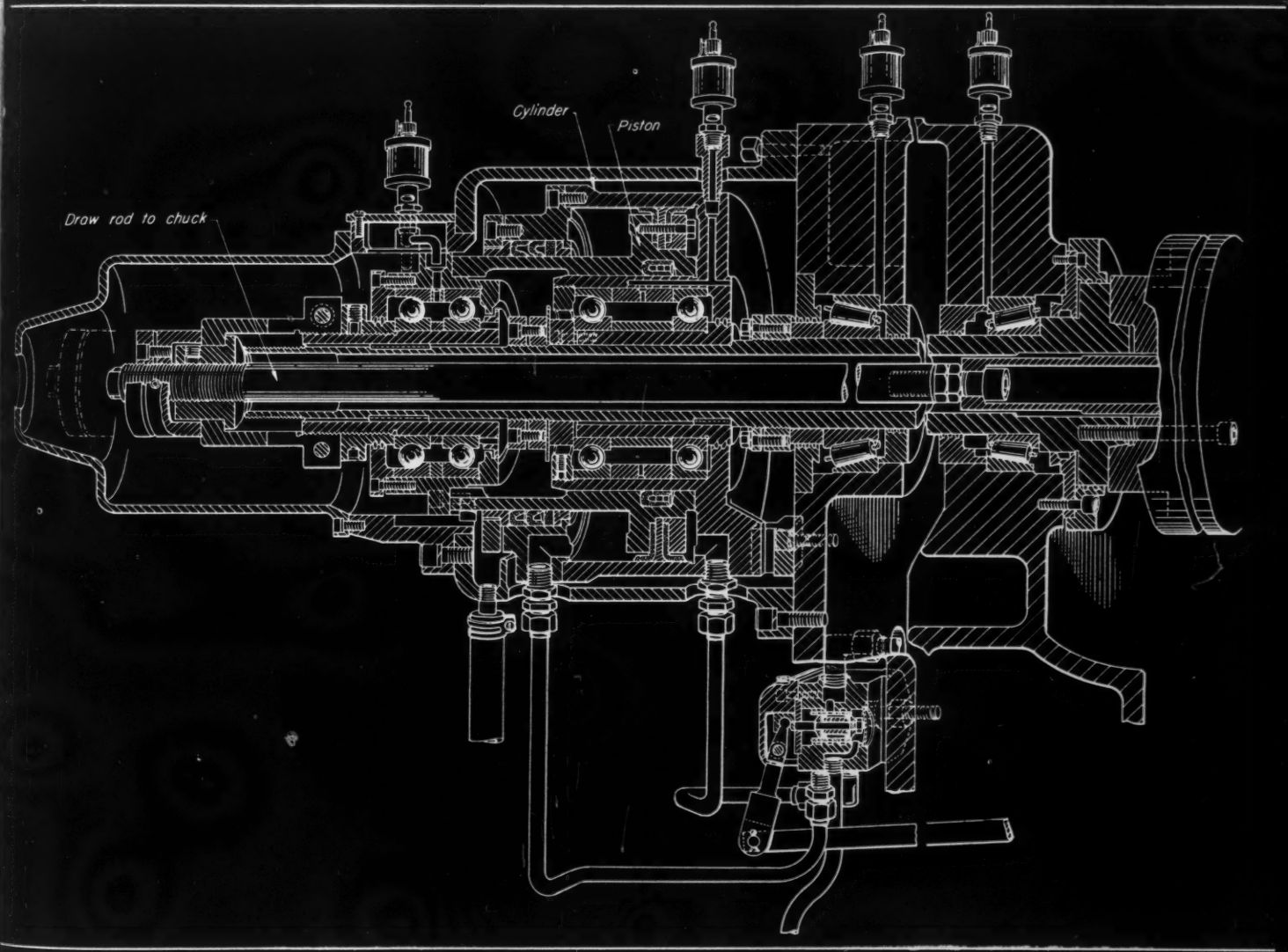


Fig. 3—Construction of low-inertia stationary air cylinder for operating the headstock chuck

volt control circuit transformer, a disconnect switch, auxiliary relays, and starters. Even though the motor is only 5/2½ hp, an oversized reversing starter (NEMA Size 2) is used to insure long life under the severe reversing service. The control cabinet is of heavy gage steel with gasketed door to exclude dirt, coolant and chips, in accordance with the Machine Tool Electrical Standards. The control is assembled and wired from standard commercial components.

Achievement of adequate performance on this Electro-Cycle turret lathe required careful attention to the inertia of machine parts and to motor design to dissipate the heat generated by the frequent starts, stops and reversals. Because the machine had to be adaptable to a wide variety of chucks already in use in customers' shops, the greatest attention was paid to reducing the inertia of machine parts. The largest single item of inertia in previous machines was in the conventional rotating air cylinder used for closing the chuck jaws. Co-operative design work with a cylinder manufacturer resulted in the development of a stationary air cylinder, *Fig.* 3. The only rotating parts are the bearings, as compared with the entire piston and cylinder of the rotating type cylinder. The inertia factors (WR^2) of the stationary cylinder and rotating cylinders are 0.9 lb-ft^2 and 6 lb-ft^2, respectively. The percentage reduction of WR^2 of a chucking machine from this feature alone is 20 per cent which provides better than a 20 per cent increase in maximum number of starts, reversals and stops per minute at 1500 rpm spindle speed.

Problems of economically providing the machine with six speeds with as low inertia as possible were solved by the use of two B-section steel-cable V-belts running on three-step sheaves, and the two-speed 5/2½ hp motor. The improved six-speed drive requires approximately the same space and has approximately the same inertia as the four-speed drive previously used. In addition to increasing the number of available speeds, the steel-cable belts give much longer service than conventional belts.

Indicates Correct Belt Tension

For belt tightening, the flange-mounted motor has a hinged cradle which is raised or lowered by a crank-operated jack screw, shown in *Fig.* 2, to the left of the door in the machine base. The cylinder in the recess above the crank socket is a calibrated spring counterbalance which provides a visual indication of correct belt tension.

Design of the motor became an important factor in the success of the machine because of the necessity for rapid functioning of the spindle. In a rapid-reversing duty cycle, the energy of the rotating parts is converted into heat in the motor rotor. To this is added the heat from the stator losses. It is necessary, therefore, that the motor design provide for maximum ventilation and heat-radiating capacity to prevent overheating. To accomplish this, the final motor design incorporated large fan blades on the rotor, long rotor bars extending beyond the rotor iron, a hollow magnesium rotor spider, and radial vent slots in the center of the rotor and stator iron. In addition, the stator is wound with Class B (glass) insulated wire, and the frame size is roughly twice the size of a normal motor of the same horsepower rating. *Fig.* 2 shows the general appearance of the motor and mounting. Use of an oversized frame also permits increasing the starting and reversing torques to 450 and 600 per cent of full-load torque, respectively. These high torques are required to give the spindle the necessary rapid response to achieve maximum production.

Brake Aids in Stopping

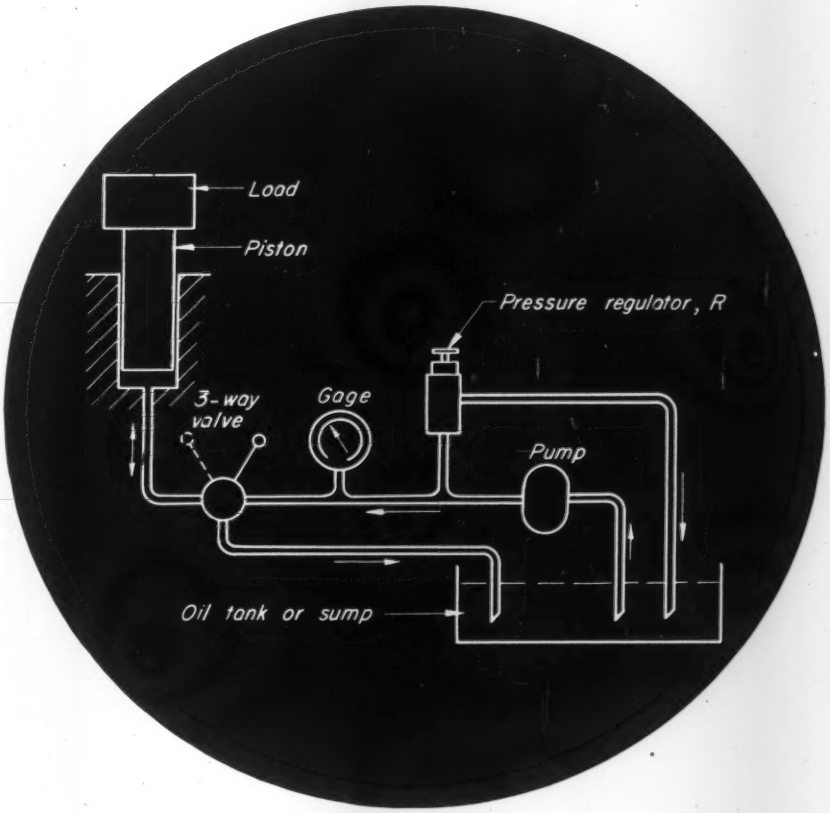
As a further aid to rapid stopping and spindle positioning, a d-c operated disk brake, originally designed for automotive trailers, is used. The setting of the brake is speeded up by a unique capacitor resistor discharge circuit that applies a momentary overvoltage when the brake is energized. Additional advantages of the brake include permanent adjustment, long life and low inertia of the rotating parts. The brake is mounted on the motor cradle within the base of the machine and the rotating disk carried by the motor sheave.

A close-differential thermostat for motor overtemperature protection is mounted on the stator and connected in the control circuit to stop the machine before the motor can be damaged. However, since the motor will cool approximately ten times faster if running idle, the circuit is arranged so the operator can run the machine in *reverse only*, by holding the power-on button depressed and holding the control lever in the reverse position. This cools the motor and resets the thermostat in a few moments, and keeps the operator at the machine to minimize lost production time.

To improve motor appearance and reduce its length, a close coupled mounting of the plugging switch was developed. This consists of a bonded rubber flexible coupling contained wholly within a bored hole in the end of the motor shaft.

Jobs involving only a few seconds can be produced at such high rates that the performance capacity of the motor may be exceeded. For those jobs, an auxiliary motor-driven blower is employed. This blower is of the axial-flow type, driven by a 1/15-hp totally enclosed motor. A sheet metal shroud encloses the center of the main motor stator and connects with a vertical stack which holds the blower. The hot air is thereby drawn out of the motor to provide more rapid cooling and is discharged upward to prevent discomfort to shop personnel. Performance capacity increases of 80 per cent are possible at some speeds with the auxiliary blower unit.

New production records are continually being set by these machines as their full capabilities are further exploited. Reports from users reveal production increases ranging from 25 to 22 per cent over previously used methods.



Selecting Hydraulic Pressure Control Valves

By L. S. Linderoth Jr.
Research Division
United Shoe Machinery Corp.
Beverly, Mass.

CORRECT selection of hydraulic pressure regulating valves requires judicious matching of the characteristics of the valves and the conditions in the hydraulic circuit of the machine. Although operating principles of such valves are well known, it is believed that many of the factors contributing to successful application are not too widely understood. To provide

designers with specific information on the performance and use of these valves, the operation and design of the more common types of safety valves, sequence valves, pressure regulators, relief valves, and pressure reducers will be discussed in this article.

A perfect pressure regulating valve will have zero regulation or "droop", no noise or vibration, and will shut off "drop-tight" below the pressure at which it is set to open. It is, of course, impossible to design a perfect regulating valve since a pressure differential is required to cause it to operate. However, it is possible to approach perfection and the means used to this end will be illustrated and explained.

Before selecting a suitable pressure regulating valve it is necessary to know under what conditions it must operate. The maximum and minimum pressure in relation to the flow of fluid through the valve must be ascertained or arbitrarily fixed.

SIMPLE HYDRAULIC SYSTEM: On the previous page is illustrated a simple hydraulic system. As long as pressure regulator R remains closed the hydraulic pressure registered on the gage is substantially equal to the load divided by the piston area. This, of course, neglects pressure drop through the pipes during flow, and piston friction, which increase the pressure of this system above this value. When the 3-way valve is operated to actuate the piston and thus to exert a force on the load, the pressure regulator must remain tightly closed so that all of the oil from the pump is available to move the piston. When the piston reaches the end of its stroke, or if the load increases sufficiently, a point is reached where the pressure in the system is sufficient to start opening the regulating valve R. The flow from the pump divides between the requirements of the piston and the excess which is pased off through the regulator and returned to the sump. When the piston motion ceases, all the pump output passes through the regulating valve and the pressure in the system is determined by the setting of this valve.

Referring to *Fig.* 1, if pressure in the hydraulic lines increases due to the piston meeting work resistance or reaching a stop then, when pressure P_1 is reached, oil starts to flow through the valve as its poppet or ball is lifted from its seat by the oil seeking an escape path. The pressure in the system increases since all of the hydraulic fluid must now flow through the safety or relief valve and, in order to do so, must lift the valve further from its seat. At flow Q in *Fig.* 1 the corresponding pressure is P_2.

REGULATION: This variation in pressure from P_1 to P_2 is known as regulation or droop. It is present to a more or less extent in all simple relief or safety valves. The flow of oil requires a lifting of the poppet which in turn compresses a spring, making it

Fig. 1—Below Left—Relation between system pressure and flow through pressure regulator valve

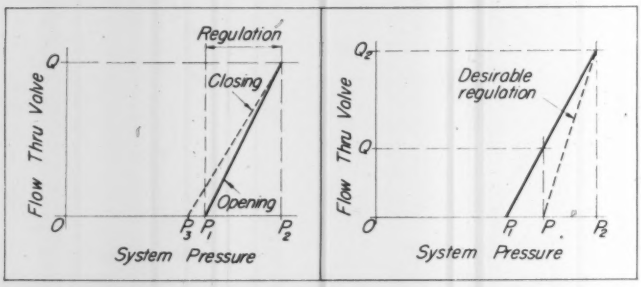


Fig. 2—Above Right—With desirable regulation there should be no flow through valve at operating pressure P

Fig. 3—Below—Simplest type of pressure-regulating valve, the ball valve is primarily a protective device

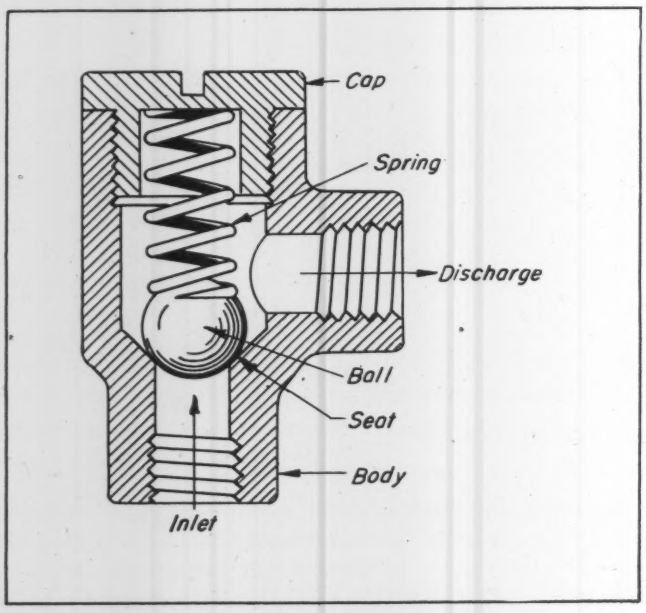
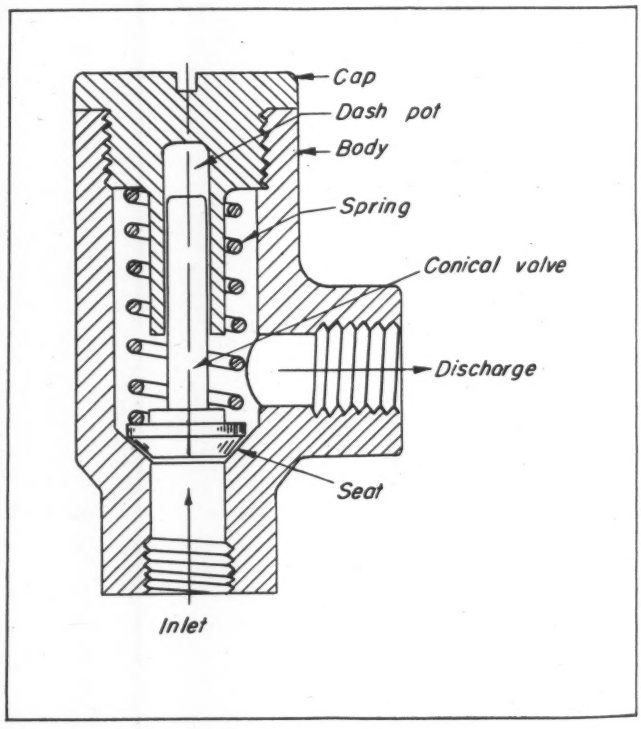


Fig. 4—Cone type valve, which is quieter in operation than the ball valve, is also used as a safety valve



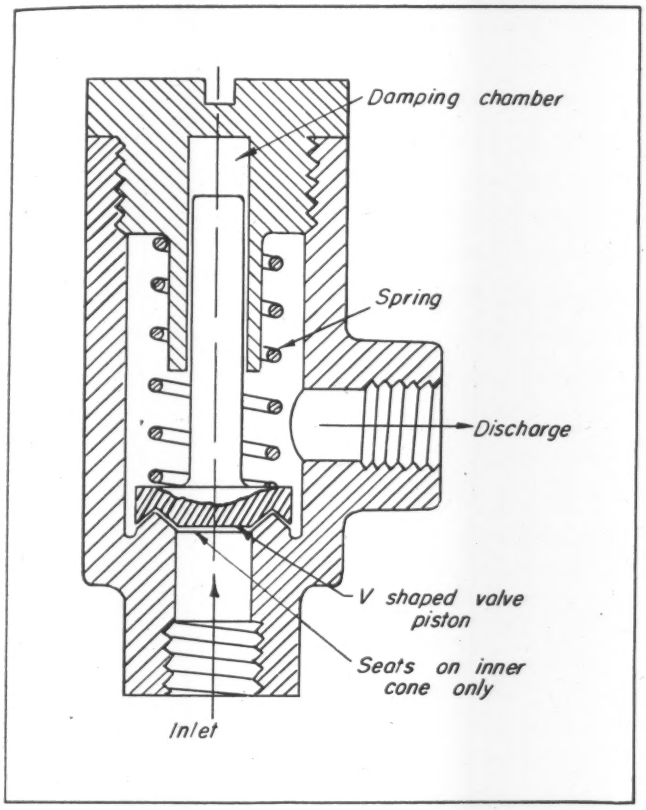


Fig. 5—Pop safety valve's effective lifting area increases when valve lifts, causing it to open more quickly

more difficult to raise the poppet, and thereby causing a pressure increase.

Referring again to *Fig.* 1, the solid curve from P_1 to P_2 represents the pressure-flow relation while the valve is opening (flow is increasing). Due to friction in the valve between the moving parts the pressure flow curve of the closing valve will not be the same, but will follow the dotted curve to a lower pressure P_3, at which point the valve will just close and permit no flow through it.

Importance of regulation is best illustrated by referring to *Fig.* 2 and the system shown on Page 105. At zero flow through the valve the pressure is P_1. The flow characteristics of the valve are indicated by the solid line from P_1. With the piston at rest and the directional control valve closed, all the oil or fluid must pass through the regulating valve and the valve operates at volume Q_2, the output of the pump, and at pressure P_2. The piston requires pressure P to move its load. When the directional control valve is opened and oil is allowed to flow to the cylinder, the pressure immediately drops to the value P and the flow through the regulating valve drops to Q. The oil reaching the cylinder is Q_2 minus Q, obviously an inefficient system since all of the oil does not have a chance to work on the piston. A change in the regulation of the valve is indicated and is shown on *Fig.* 2 by the dashed line. Now Q has been reduced to zero at pressure P, and all of Q_2 goes to work on the piston.

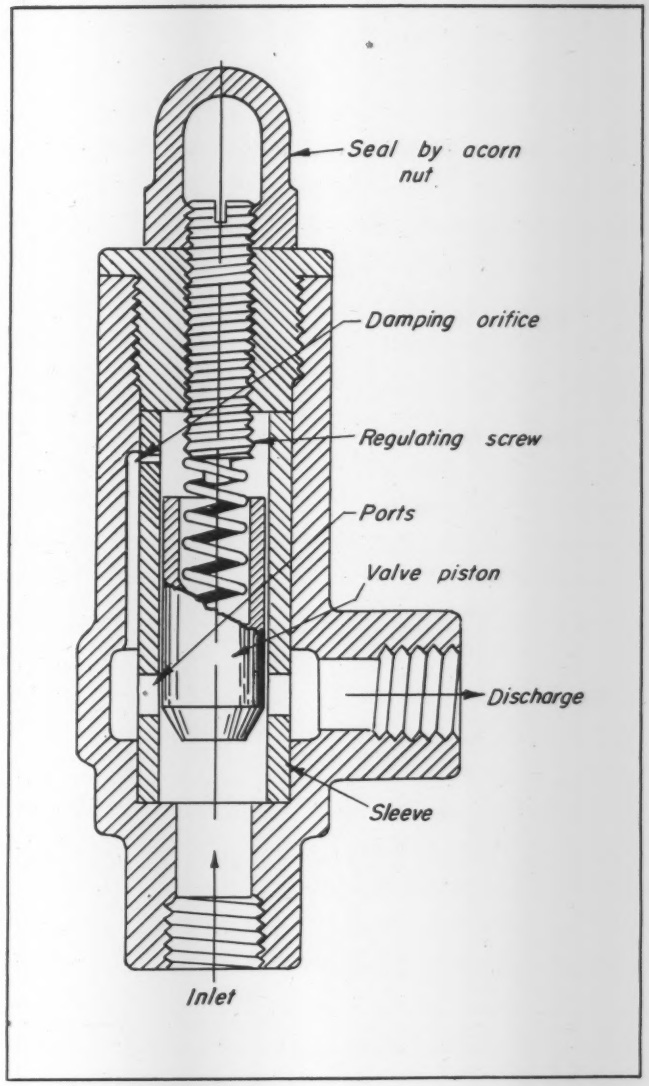


Fig. 6—Pressure-regulating valve for continuous operation. Operation is quieter than ball or cone types

BALL TYPE VALVE: In *Fig.* 3 is illustrated the simplest of all pressure regulating valves. It is more accurately called a pressure safety valve, since its greatest use is as an overload relief. It consists essentially of a housing or valve body in which the oil enters through a port having a sharp-edged or a conical seat against which a spherical ball is held by spring pressure. The spring load may be fixed or adjustable. The oil, when sufficient pressure builds up, raises the ball off its seat and passes around it to the outlet port.

These valves are excellent as protective devices to prevent the rupture of hydraulic elements due to excess pressure. This design is not adequate for continuous duty as a pressure-regulating means, except for very small flow. It tends to be noisy in operation and has poor regulation.

CONE VALVE: An improvement over the ball is illustrated in *Fig.* 4. Here the ball is replaced by a conical plug or valve. The method of guiding the

Fig. 7—Right—Zero-regulation valve employing weight instead of spring. Operation would be unstable

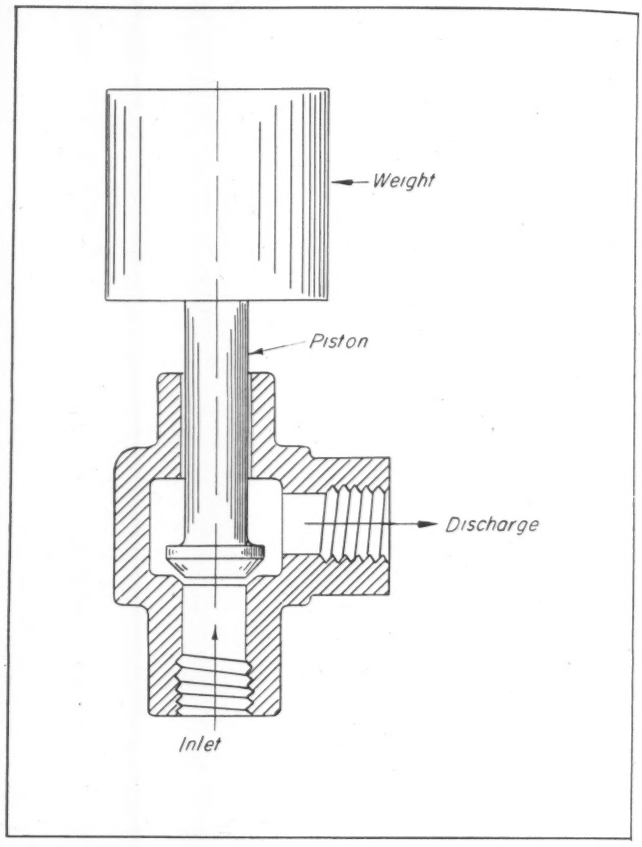


Fig. 8—Below—Close approximation to zero regulation results from use of low-scale spring and small lift

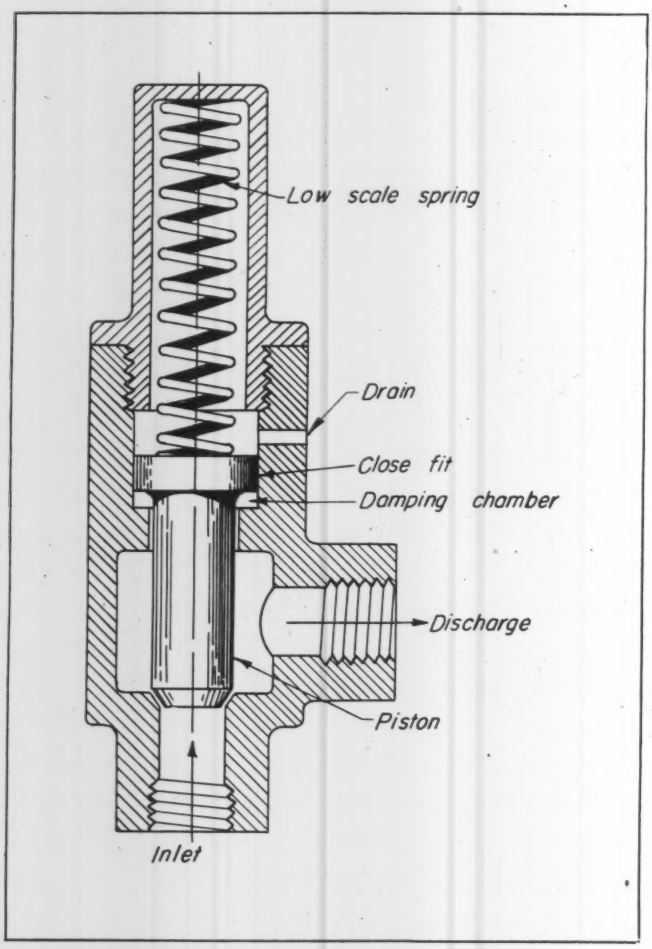
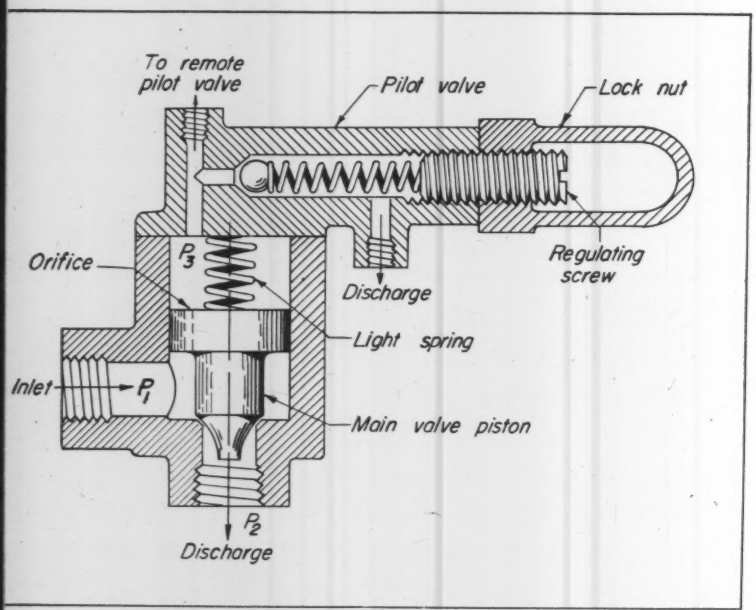


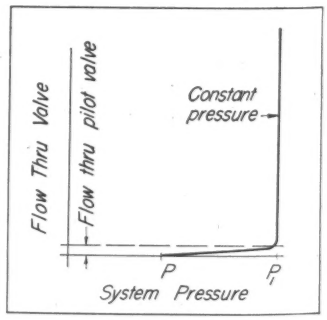
Fig. 9—Below—Hydraulic piston in place of spring to control valve lift gives regulation nearly zero



conical plug may vary widely, but the principal objective of holding the plug central with its seat and parallel to it must be achieved for satisfactory operation. The regulation of this design is essentially the same as for the ball type. These valves are used principally as safety valves. They are quieter in operation than the ball valves since the damping introduced by the guide stem reduces the vibration prevalent with the ball.

POP SAFETY VALVES: A variation of the foregoing type of valve is found in the so-called "pop" safety valve. The principle of the design is illustrated in *Fig.* 5 and consists essentially of a means for increasing the effective lifting area of the valve, so that as soon as the valve starts to open the fluid reacts on a greater area and thereby exerts a greater force on the valve plug or stem, opening it more quickly and causing it to "pop". These valves are commonly used as safety valves on steam boilers.

Fig. 10 — Pressure-flow diagram for the valve shown in Fig. 9, illustrating the closeness of the regulation



CONTINUOUSLY OPERATING VALVES: In *Fig.* 6 is illustrated a simple valve designed primarily for continuous operation as a pressure regulator in a hydraulic circuit such as is shown on Page 105. The oil flow is regulated by allowing it to lift a spool to open ports in a sleeve, allowing the oil to pass through these ports into a chamber and then to the outlet. The motion of the spool is damped by the friction against the sleeve and by restricting the flow of oil from above the piston. In general, the operation of this type of valve is quiet. Its regulation is a function of the spring characteristics in relation to the port openings.

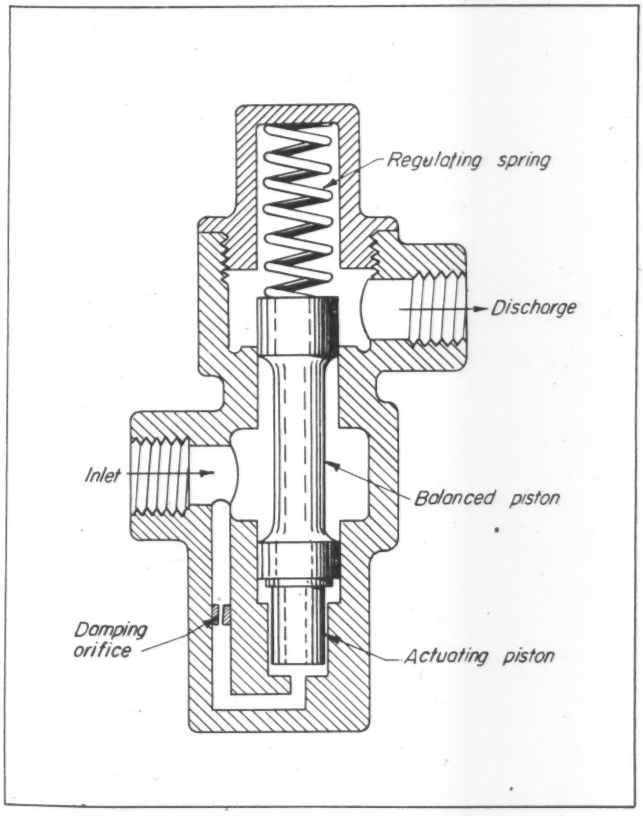
ZERO REGULATION: It is possible to design valves in which the regulation is substantially zero, such a valve being illustrated in *Fig.* 7. A weight is used to load the piston so that the force against which the oil must work to open the valve remains constant regardless of valve lift.

In practice the weight is replaced by a spring with a very low scale and the area of opening is made large for a very small lift. This design would be unstable unless suitable damping were incorporated. A valve of this type is illustrated in *Fig.* 8.

A valve of nearly zero regulation can also be made using a hydraulic piston in place of the spring, and regulating the pressure on it by using a small ball type regulator in a pilot circuit. Such an arrangement is illustrated in *Fig.* 9. The principle of operation is to balance the pressures on either side of the operating piston in such a relation that the valve is lifted off its seat the proper amount to give the required pressure P_2. An orifice bleeds off a sufficient quantity of oil to give the valve adequate response. This oil is controlled by a pilot pressure regulator which can be of relatively simple design since its characteristics do not appreciably affect the main valve. When the valve is in balance, the flow Q establishes a height of the valve off its seat.

If a higher pressure is required, the pressure P_3 is raised, causing the main piston to close the valve and

Fig. 11—Balanced spool actuated by auxiliary piston permits reducing size of spring for high pressures



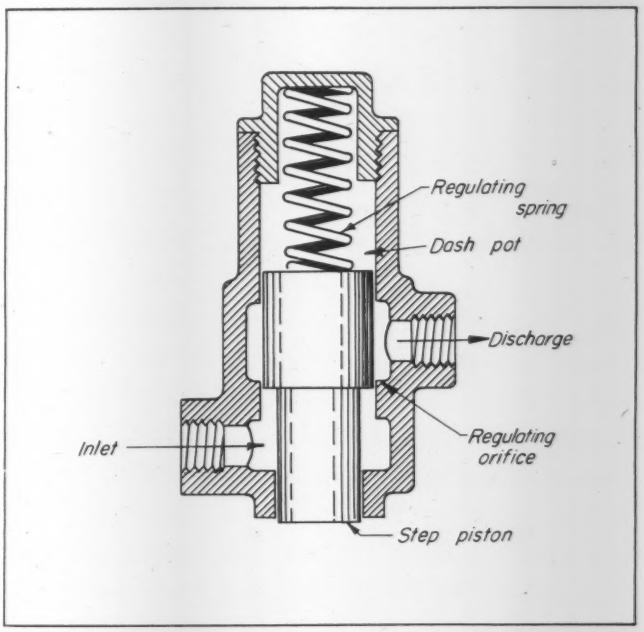
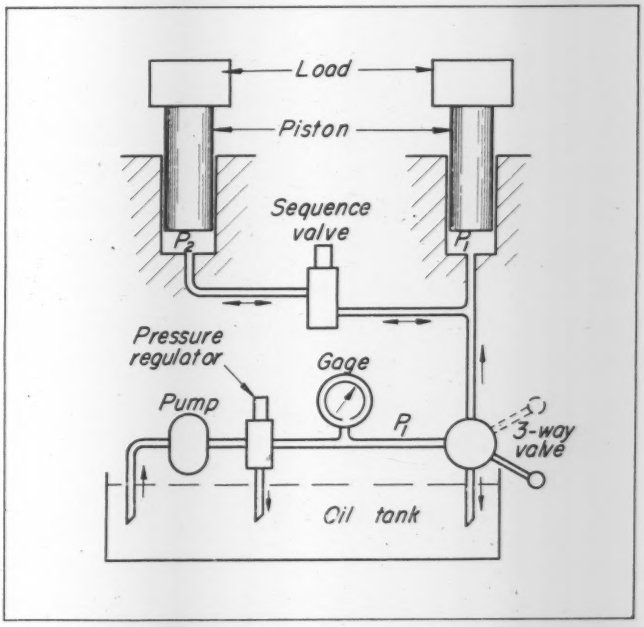


Fig. 12—Above—Hydraulic pressure on step of small-area piston is easily balanced by moderate-size spring

Fig. 13—Below—Circuit employing a valve of the type shown in Fig. 12 as a sequence valve



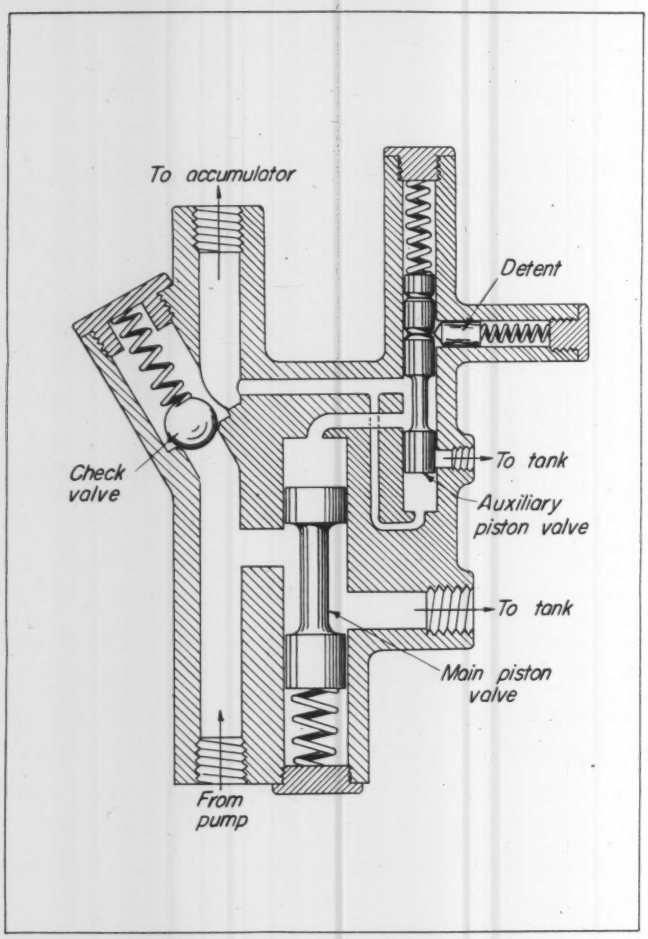
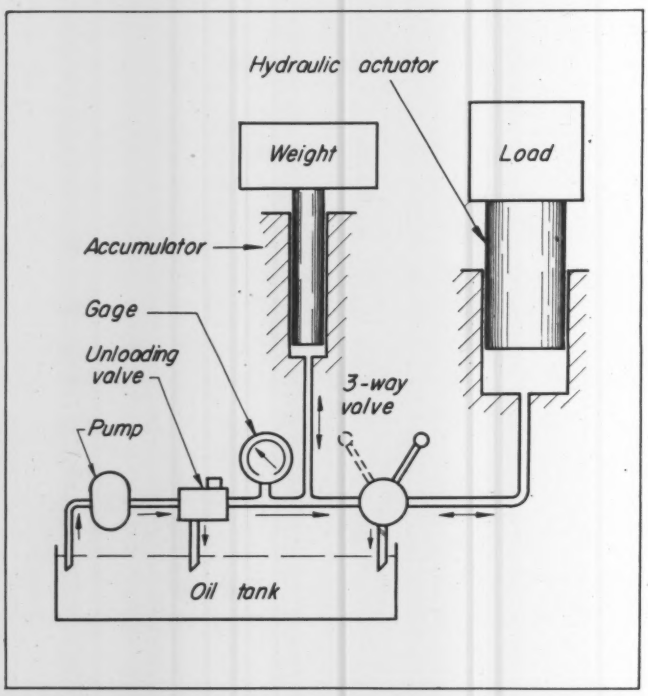


Fig. 14—Above—Unloading valve shown in unloaded position, which is maintained by pressure from accumulator

Fig. 15—Below—Typical circuit using unloading valve to relieve pressure on pump when not required



thereby raising pressure P_1 until a balance of the main piston is again reached. If the flow Q is diminished, the pressure P_1 drops, causing the piston to close the valve until pressure P_1 is again reached to maintain equilibrium. This action is automatically damped by the orifice and the valve assumes a state of steady eqilibrium. Back pressure P_2 will affect this valve adversely. The pressure-flow diagram for this type of valve is illustrated in *Fig.* 10.

SPECIAL TYPES: For higher pressure, where it is desirable to keep the overall size of the valve small, it is possible to reduce the size of the spring by cutting down the lifting area of the spool against which the hydraulic fluid acts. This may be done as in *Fig.* 11, where the spool or piston is hydraulically balanced and is actuated by an auxiliary piston, or as in *Fig.* 12, by the use of a stepped or two-diameter piston. The reduced area of the auxiliary piston or the annulus of the stepped piston result in smaller forces for the spring to act against.

ANALYSIS OF FLOW THROUGH VALVE: All of the valves illustrated and described so far, with the exception of *Fig.* 9, have pressure characteristics similar to that shown in *Fig.* 1. The pressure required to maintain a given flow through these valves is considerably affected by the back pressure in the outlet against which the fluid must flow.

Flow through the orifice of a relief or safety valve is turbulent, hence the equation for this flow can be written

$$Q = K \times A \times \sqrt{P_1 - P_2} \quad \ldots\ldots (1)$$

in which Q = flow (gallons per minute) through the valve; K = constant for valve under consideration; A = orifice area (square inches); P_1 = inlet pressure (psi); and P_2 = outlet pressure (psi). The usual dimensions used for these components are in parentheses. The constant K will include the factors to make the equation dimensionally correct. It is also possible for the constant K to vary slightly due to the change in orifice shape as a result of the motion in the valve piston caused by a change in the radical $P_1 - P_2$. For all practical purposes this change can be neglected.

It is evident from Equation 1 that a change in the flow Q must result in a change in either A, P_1 or P_2 or a combination of these components. Assuming for the moment that the valve shown in *Fig.* 4 is being considered and that P_2 is zero, Equation 1 then becomes

$$Q = K \times A \times \sqrt{P_1} \quad \ldots\ldots (2)$$

In this valve, *Fig.* 4, as Q increases P_1 increases. This causes a force to react against the valve spring and to lift the valve further off its seat, increasing A. Equilibrium is reached when Equation 2 is again satisfied.

If the change in flow is abrupt, the motion of the valve piston may be violent enough to cause it to overtravel, thereby increasing the area A to a larger value than required, which in turn causes a reduc-

tion in P_1. A lower value of P_1 causes the spring to return the valve piston violently and a forced vibration is set up. This vibration can be damped by various means to bring the valve piston quickly to a steady state. Friction or dash pots or a combination of the two are commonly used. This is illustrated by the valves in *Figs.* 6 and 8. The valve in *Fig.* 4 depends on friction of the valve stem in its guide and the dash pot action of the blind hole in the guide. The ball valve in *Fig.* 3 is practically undamped.

EFFECT OF BACK PRESSURE: Equation 1 also shows the effect of back pressure P_2 on a valve. If the flow Q remains constant and the back pressure P_2 increases, two things may happen to satisfy the equation: A may increase or P_1 may increase an amount equal to the increase of P_2; or both A and P_1 may change.

Response to Back Pressure

Using again the valve in *Fig.* 4, an increase in P_2 will cause a greater down load on the valve piston and this will have to be offset by an increase in P_1 to keep the piston open to area A. Conversely, if P_2 is decreased P_1 will decrease a like amount. It will be noticed that the back pressure P_2 in the valve in *Fig.* 4 acts on the entire valve stem cross section as well as the conical head.

Referring to *Fig.* 7, the dead weight loaded valve, and assuming that the valve stem area is the same as the conical head area on which pressure P_1 acts, the following conditions exist in satisfying Equation 1. The back pressure P_2 has no effect on the forces moving the valve stem. As P_2 increases or decreases A does the same in order to satisfy the equation. In a spring-loaded valve, A could not change without changing P_1, so that if P_2 is varied in a valve such as *Fig.* 12 there is a change in P_1 of an amount depending on the characteristics of the spring. With a low scale spring and proper valve piston design, this change in P_1 with a change in P_2 can be made substantially zero. Valves of this type are frequently used as sequence valves to insure that the pressure P_1 is reached in the primary circuit before pressure P_2 starts to build up in a secondary circuit. This is illustrated by the system in *Fig.* 13.

UNLOADING VALVES: Another method of pressure regulation is found in the pressure unloading valve. This valve is used in conjunction with an accumulator and serves to unload the pump during the time when a flow of oil under pressure is not required. The basic features of this type of valve are illustrated in *Fig.* 14.

Operation of the unloading valve in *Fig.* 14 is cyclic or intermittent. When no pressure exists in the system, the main valve closes the port connecting the pump to the tank return line. Oil or fluid from the pump is forced to take a path through the check valve and into the accumulator, building up pressure in this line. This pressure is communicated to the auxiliary piston valve by a suitable connecting pipe or duct in the valve housing. When a pressure suf-

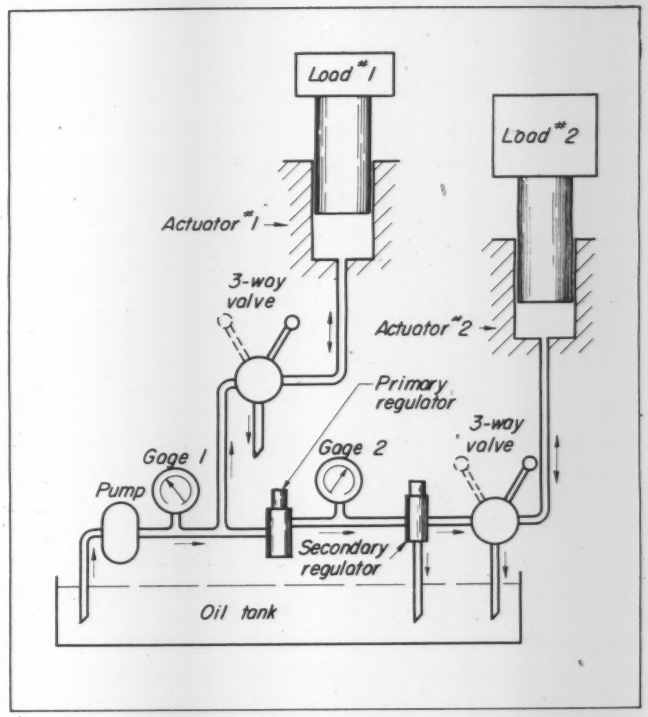
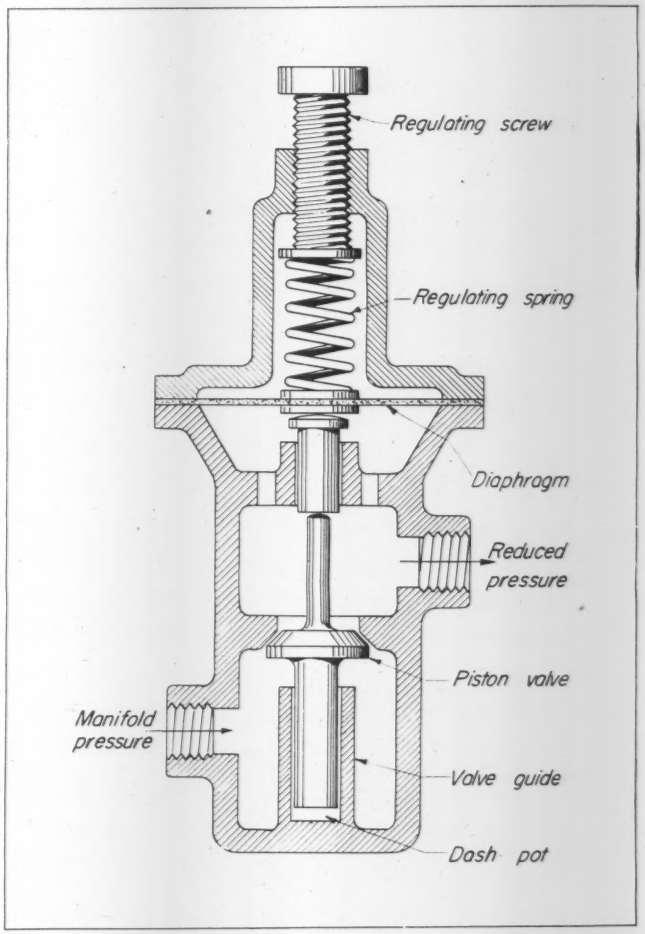


Fig. 16—Above—Circuit for multiple-pressure system in which one pressure is always larger than the other

Fig. 17—Below—Reducing valve shuts off flow when pressure on downstream side attains predetermined value



ficient to overcome the opposing spring and detent force of the auxiliary valve has been reached the valve will quickly move up to the position shown in the sketch, *Fig.* 14, allowing this pressure to communicate with the main valve and causing it to move very rapidly to a position which allows the entire flow from the pump to return to the tank at substantially zero pressure. At the same instant the pump pressure is dropped by the motion of the main valve, the check valve closes and prevents flow from the accumulator to the tank.

The valve will remain in the "unloaded" position as long as sufficient pressure exists in the accumulator to hold the pilot piston valve against the action of its spring. As soon as this pressure drops to a point where the spring can overcome it plus the detent resistance, the auxiliary piston valve will snap down into the "load" position and allow fluid holding the main valve open to escape to the tank, allowing the main valve to close the return to tank port. The pump pressure then rises to overcome the accumulator pressure on the check valve and then refills the accumulator through the check valve. When pressure is re-established the cycle repeats itself as long as the pump runs and fluid is used or lost from the accumulator. A typical unloading valve and accumulator circuit is illustrated in *Fig.* 15.

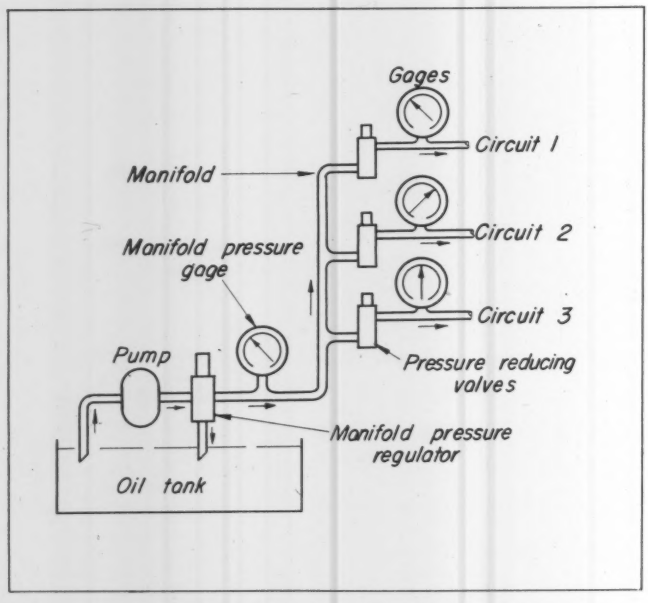


Fig. 18—Typical hydraulic circuit employing pressure reducing valves of the type shown in Fig. 17

The unloading valve and accumulator are used when the hydraulic motor only operates a very small fraction of the entire machine cycle and when the pressure and maximum flow are not needed during the remaining part of the cycle. The accumulator serves as a storage reservoir for the energy needed to perform the required work in the motor, and the pump need only be large enough to supply this energy over the entire cycle, whereas the accumulator may be called upon to deliver this energy in a very small fraction of this cycle. Since power is energy divided by time it can readily be seen that this type of system may be a means of saving a great deal of power.

MULTIPLE-PRESSURE SYSTEMS: It is frequently desirable to use two or more different and variable pressures in a hydraulic circuit. If the one pressure is always larger than the other one, a sequence valve such as in *Figs.* 8 or 12 can be used, together with a pressure regulator as in *Figs.* 6 or 9 to form the circuit illustrated in *Fig.* 16. As long as the pressure on gage 2, *Fig.* 16, is less than gage 1 the controls on either pressure are independent.

Regulator plus Reducing Valve

When it becomes necessary to vary either pressure to values higher or lower than any other pressure a reducing valve is necessary. A manifold pressure equal to the highest pressure required is maintained and regulated by a valve as in *Figs.* 6 or 9. Lower pressures are regulated by the reducing valve illustrated in *Fig.* 17.

The valve is so designed that the back pressure on the piston (the regulated pressure) causes the valve to close pressure tight when the desired pressure is reached. Fluid enters the valve from the manifold. If the pressure on the downstream side of the valve is low the force of the regulating spring is great enough to overcome this downstream pressure and open the main valve piston. This allows fluid to pass through the valve. As the pressure is increased by the fluid flow this pressure communicates to the bottom of the diaphragm and raises it against the regulating spring force. This shuts off the flow of fluid, thus preventing further pressure increase. As the pressure drops and more fluid is needed the spring again opens the main valve. If a steady flow is required the valve will remain open at a point that will give the correct pressure drop for this flow to give the desired reduced pressure. This valve is shown in *Fig.* 18 in a typical hydraulic circuit. Any number of these valves may be operated from a single manifold, provided no valve or combination takes more than the available flow of fluid.

CONCLUSION: It should be pointed out that none of the valve sketches is to scale or correctly proportioned. These sketches were drawn primarily to illustrate the principle of operation or type of design used in each instance.

There is considerable confusion regarding the real or fancied difference between safety, relief, and pressure regulating valves. Actually, as has been shown, any valve that has a means for adjusting the pressure at which it will open can be called a pressure regulator, if by regulation is meant the adjustment of pressure. Likewise any valve that will open at a set pressure may be called a safety or a relief valve. The use to which a valve is put should determine what to call it.

Engineers' Symposium

Drive and control features utilized in machines are presented in this group of short articles

Adjustable Drive Utilizes Flexible Shafts

By George E. Ryckman
President
Ryckman Machine Co. Ltd.
Stoney Creek, Ontario

In decorating china ware in the pottery business, one of the methods is to apply an edge line to the ware. In this particular instance the edge line is applied by a wide-faced soft-rubber roller and the width of line is determined by the contact condition prevailing.

In developing the design of this machine the following conditions were to be met:

1. Contact pressure was to be adjustable from one ounce to fourteen ounces.
2. Rollers were to be quickly interchangeable and the set up was to be arranged to allow the use of any roller size between one-half inch diameter and two-inch diameter, without changing peripheral velocity.
3. Applicator heads were to be power driven so that thick colors could be used.
4. Applicators were to be universally adjustable so that any size or shape of pottery could be worked.
5. Applicators were to be arranged in pairs so that balance loading on the pottery ware would simplify chucking.
6. Opposite rollers of the pair (at each station) were to rotate in opposite directions because of their relation to the pottery ware item.

Solution of the problem as embodied in the machine illustrated consists of a circular continuous-rotation table with six work heads equally spaced about the table. A fixed nonrotating shaft is brought up through the main table and to this is fixed a large driving spur gear. Meshing with this central driving gear are six pairs of driven spurs secured in a cage and rotating with the table. As these driven pinions ride with the table in a circular path about the fixed spur gear they rotate about their own axes. The drive motion is from each driven pinion to an applicator head by means of a flexible drive shaft which is restrained somewhat where it bears in the universal joint connection through which the machine arms are set for size of product. The flexible drive shaft passes through the tubular arm and to the applicator head through a spline fitting which allows approximately one and one-half inches of axial movement.

Since the applicator heads are mounted on tubular arms the applicator may be rotated about the axis of the tubular arm without creating a drive problem. The contact pressure of the roller on the work is adjusted by counterbalancing the undesired portion of the applicator weight by means of a tension spring in an adjusting guide.

The condition of constant peripheral velocity for all sizes of rollers is obtained by making the applicator roller an idler, running against a constant velocity drum from which the applicator roller also obtains its color at each revolution. Since it is also desirable to have the applicator arms swing clear of the work as the work station passes the operator, the arms are arranged to swing vertically while the flexible drift shaft continues to rotate inside of the tubular arm.

The vertical arm motion is obtained with push rods working on a pair of horizontal cams attached direct to the central nonrotating spindle. The length of the push rods is adjusted by means of a threaded nose piece having fiber friction faces to assure constancy of settings during vibration.

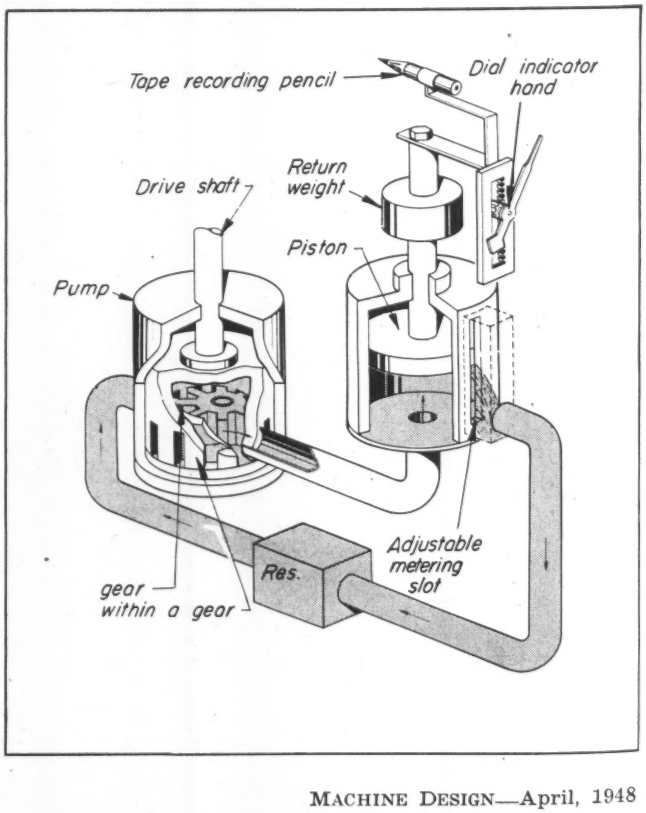
# Indicates Speed from Hydraulic Circuit

By J. A. Roberts
Engineer Chicago Pneumatic Tool Co.
New York

IN DESIGNING a speed recorder for railway service, the most essential problems involved the stability and accuracy of the indicating and recording mechanism. Obviously if visual indications and written records were to be of any value to railroad engineers and operating departments they would have to be continuously reliable.

It was thought that a completely closed hydraulic system rather than a spring tension system would be the most consistent and stable medium to reflect locomotive movement, and the designers' efforts were directed toward that end. After several years of research and experimentation the CP-75 and CP-120 railway speed recorders were perfected, the basis of which is the pump and piston arrangement discussed in the following.

The impelling unit of the hydraulic system is a modified Viking positive-displacement rotary pump, driven direct from the journal box or locomotive driver through a gear-reduced flexible shaft. This particular pump was selected because it pumps a con-



tinuous steady stream, free from pulsations, and is so simple in operation that it is not damaged by vibration or temperature change.

Since the displacement is positive its output varies in direct proportion to its speed. The closed circuit, between the pump and the piston it motivates, obviates the inherent weaknesses of suction or lift characteristic of other types of mechanisms. The hydraulic pump operates at very low speed which means that the load on the flexible shaft is not excessive.

In the speed recorder a hand on an illuminated dial indicates to the engineer the speed of the train, and a pencil records a log of a complete trip on a calibrated strip of tape. Both of these mechanisms are actuated by a free-fitting piston mounted in a cylinder.

The cycle is as follows: The wheel-driven flexible shaft drives the low-speed pump. This pump, completely submerged in fluid delivers fluid into the cylinder chamber beneath the piston. When the locomotive increases speed, it increases the output of the pump which increases the volume in the cylinder chamber. As the volume increases the piston is forced upward, uncovering more of a discharge orifice, proportional to the pump output.

When the speed is reduced the piston descends to cover more of the discharge orifice in proportion to the reduced discharge of the pump. The cylinder wall is slit with a narrow discharge orifice. As the piston is forced upward the amount of discharge orifice area is automatically increased. This area is so designed that it exactly accommodates the output of the pump, at high, low or intermediate speeds.

The recording pencil is attached to the piston, and traces every up and down movement. A rack and pinion arrangement revolves the visual indicator hand with the same degree of accuracy.

It will be seen that throughout the whole operation, action is positive and is governed solely by the stable laws of hydraulics and gravity. One of the main reasons for adapting a hydraulic system was to avoid the use of springs throughout the cycle of operation. The only spring used is to hold the recording pencil against the tape. It is manifest that where permanent accuracy is involved, the mechanical vagaries of tension springs cannot be tolerated.

Calibration of the instrument is easily effected without excessive dismantling, through an ingenious use of the discharge orifice. Two external regulating screws widen or narrow this orifice so that the capacity of the slot uncovered by the piston equals the output of the hydraulic pump at any speed throughout the range of the instrument. When this condition exists, determined by a synchronous motor calibrating machine, the instrument is perfectly calibrated. Because of the closed hydraulic circuit, the machine need rarely be recalibrated.

The chart registration roll is worthy of mention. A paper chart is simultaneously printed and perforated by a machine designed expressly for this purpose. The perforations actually constitute a continuous rack, by means of which the chart is mechanically geared to the registration roll. Thus a positive feeding mechanism is provided as the pencil records the performance of the locomotive on the chart.

Differential Registers Synchronous Speed

By J. H. Gepfert
Reeves Pulley Co.
Cleveland

MANY industrial processes require tandem operation of two or more machines to form a sequence of operations. Installations of this type must have a means for setting the speed relationship between the machines to duplicate or to have an exact difference in speeds, in order to eliminate a stacking operation between the machines. A system for handling a problem of this type provides the necessary indication and adjustment between a roughing and finishing wood planer.

As may be seen in the sketch, a chain connection is made between the feed rolls of the roughing planer and one shaft of a 3-shaft differential gearbox. A second shaft of the differential is connected to the feed rolls of the finishing planer the rolls being driven by a variable-speed unit.

Internal connections and drive relationships of differential are so arranged that when the variable-speed drive of the finishing planer is adjusted to give exactly the same speed as the feed rolls on the roughing planer, the third shaft of the differential remains stationary. A pointer attached to this shaft would assume, for example, a vertical position for synchronism. A difference in speed between the two planers causes the pointer to rotate at a speed which is one-half the difference in speed between the two shafts, and in the direction of the shaft which has the highest speed.

The operator can adjust the speed of the finishing planer through the variable-speed drive to obtain synchronism between the feed rolls by making whatever correction is necessary to bring the indicating pointer to a standstill. A dial mounted behind the pointer indicates if an increase or decrease in speed is required.

For processes which require a fixed difference

in speed between the two machines, the ratios of the connecting chain drives from the machines to the differentials are chosen to give equal speeds at the two input shafts of the differential for the desired speed relationship. The operation of the pointer is the same as for synchronous drives since the indicating shaft rotates on the basis of the difference in speed between the two shafts of the differential.

A further refinement of this system can be made by using a small variable-speed drive between one shaft of the differential and one of the feed rolls. This allows the operator to select, while the system is in operation, a speed difference which he finds desirable at any current condition of operation of the feed rolls.

Primary value of the differential indicator and variable-speed drive system is the simplicity of indication and adjustment for selecting an exact speed at any time so the operator can maintain synchronism or an exact speed difference between two sections or two machines. When gear connection is used, the operator has no means for setting the speed differences between the two machines, which might be required in order to process the particular feed satisfactorily.

The system described here is a means of giving the operator complete control over the speed of material between two machines, and the adjustment necessary for taking care of the slight differences in processing conditions which occur continuously in handling any type of material.

Assures Maximum Braking Effort

By C. F. Hammer
Engineering Manager
Westinghouse Air Brake Co.
Wilmerding, Pa.

THERE is nothing that rolls like a ball or an unbraked wheel on a smooth steel rail. Conversely, there is nothing that can stop rolling so easily as a heavily braked wheel on a slippery rail. When wheels stop rolling and start to slide, flat spots are worn on the wheel tread—the longer the slide, the larger the spot. These flat spots produce a pounding sound as the wheel revolves and if the spot is sufficiently large, set up vibrations that are carried upward into the car.

From the time brakes were first applied to car wheels, operating and maintenance men have been greatly concerned over the problem of flat wheels. In addition to the annoyance and discomfort to passengers, the sliding of wheels necessitates the removal of the car from revenue service while the wheels are being replaced. Then, too, wheel life is greatly reduced because the damaged wheels must be machined to remove the flat spots and any burned metal underneath the spot.

As train speeds became higher, braking forces had to increase to maintain reasonably short stopping distances. Fast service and emergency stops still had to be made on any and all rail conditions. The number of flat spots was bound to increase unless some devices were produced to prevent slides. One means employed is a speed-governor device to automatically lessen the braking force as the train speed is reduced. This was effective for average conditions, when the rail surface was reasonably dry and clean, but did not insure freedom from sliding on occasions when the adhesion between the wheel and the rail had been reduced by uncontrollable circumstances to a point where even moderate braking pressures might cause wheel slippage. Another supplementary brake control system was therefore produced to guard the wheels against sliding under all adverse rail conditions.

In order effectively to prevent sliding, the wheel protection device must be capable of distinguishing between normal retardation rates of the wheel and the rate when the wheel is slipping prior to sliding. It must anticipate a slide and function during the early part of the slipping period. Otherwise, the rapidly decelarating wheel will have reached the sliding stage before the braking force that produced the slippage can be relieved. The reaction time must be only a fraction of a second or the wheel may slide with the brakes still applied. The equipment designed to fulfill these requirements is known as the "AP" Decelostat equipment.

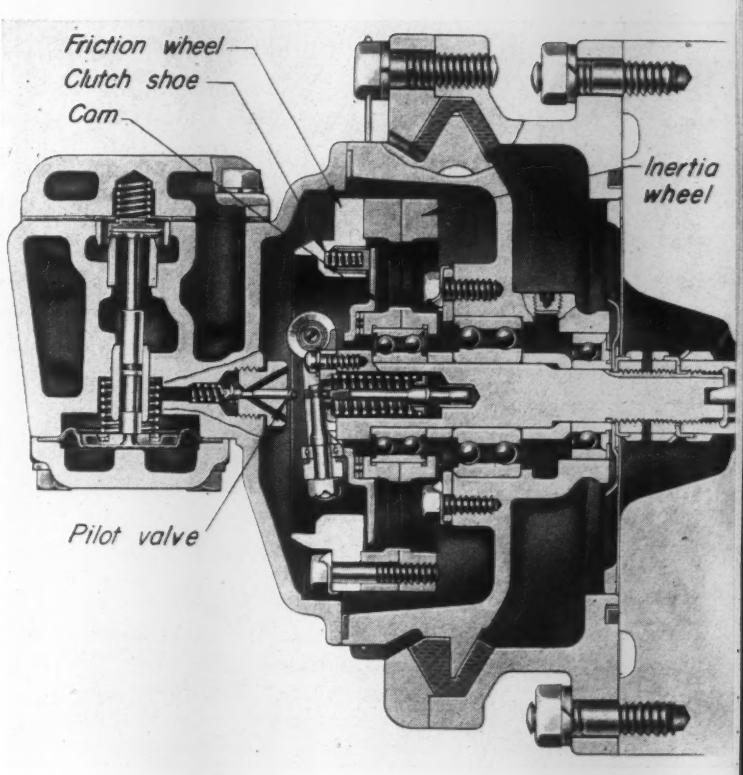
The equipment consists essentially of two operating portions, namely, an air controlling valve and a wheel retardation measuring device called the Decelostat. The valve is mounted on a truck member and the measuring device is attached to the

journal box cover as shown at right. The valve is interposed in the pipe leading from the regular brake control valve to the truck mounted brake cylinders. This valve also has an air pipe connection to the Decelostat.

The Decelostat is an inertia weight type of device, the flywheel portion of which is driven by a spline-ended shaft that extends into a splined plug in the end of the wheel axle. The Decelostat is supported by a vibration absorbing rubber ring and is held in place in the bracket by a clamping ring. The bracket is mounted to the flange on the front cover of the journal box. The pilot valve body, which contains the pneumatic venting and timing portions, is attached to the mechanical portion which in turn contains the rotating, regulating and actuating parts.

Construction of the Decelostat is shown at right. The drive shaft is supported on a front double-race ball bearing and a rear single-race ball bearing. An inertia wheel is ball bearing mounted on the drive shaft and is bolted to the friction wheel. The cam rotates on the drive shaft and is held in driving engagement to the friction wheel by three clutch shoes. The cam lever, supporting the cam shaft with cam roller, is hinged to the end of the drive shaft and the end of the cam shaft is yoked through a pull rod to a spring which holds the roller on the cam under a predetermined tension.

When the car is in motion the drive shaft rotates the cam lever and cam roller, the latter driving the cam. The cam in turn drives the inertia wheel by friction engagement of the clutch shoes. The wheel is thus driven at the same rate as the drive shaft as long as the car axle is turning and not retarded beyond a predetermined rate limit. When a brake application is made and the axle is retarded, the drive shaft is also retarded at the same rate. The inertia wheel tends to rotate at the original speed on its ball bearing and overtravels the drive shaft. If drive shaft retardation is at, or below, a normal rate the inertia force of the friction driven inertia wheel is balanced by the force of the spring. The inertia wheel thus causes the cam roller to retain a position corresponding to the initial retardation rate. If the drive shaft retardation exceeds a predetermined value, the inertia wheel overtravels and it slips on the driving clutch and causes the cam roller to move up on the cam, raising the cam shaft and pull rod to a position where the pull rod engages and unseats the pilot valve. This action operates the relay piston to vent the hose and pipe leading to the valve. Brake cylinder air pressure on that truck is quickly reduced for a brief period. When the slipping has ceased, pressure is again automatically restored. If the adverse rail adhesion that produced the slippage has not been passed or improved by means of sand, the venting and reapplication cycle will be repeated as called for by the slipping wheel. The overall train retardation is thus maintained at the maximum level possible as compared with that which would occur if the wheel sliding actually developed.



Various Strokes and Speeds from Common Shaft

By L. H. Browne
Chief Engineer
American Brake Shoe Co.
Westport, Conn.

TRENDS toward higher operating speeds have resulted, in most cases, in improved design, less weight, lower manufacturing costs, and more eye appeal, without loss in mechanical efficiency. There are many types of machines, however, that cannot be operated at motor speeds without serious drop in efficiency, such machines are usually of the reciprocating displacement variety. Designers of fluid pumps, air compressors and similar equipment have a real problem in following the trend when compelled to retain some conventional means of speed reduction such as belts, gearing, or chains. Assembled

on some kind of base or platform it may be a very efficient machine but definitely lacks the "new look."

The epicyclic eccentric drive unit illustrated was developed primarily to eliminate external speed reducing means and to enable the motor shaft or high speed driving shaft to be direct connected to the eccentric unit, producing the desired reciprocating speed and stroke.

Several pieces of equipment have been tested on two and three horsepower 1800 rpm motors, with a speed ratio of 4-1 and 6-1. All tests to date have indicated higher mechanical efficiency than conventional gearing or belting. Any ratio between the driving shaft and the driven eccentric, within the limits of epicyclic gearing, can be obtained when the eccentric contains an epicyclic gear train.

In its simple form the eccentric unit consists of four component parts—outer sleeve or ring gear, eccentric, drive pinion, and planet gear.

The speed reduction of the unit is governed by the epicyclic gear formula, where the ratio is $S/(S+R)$, S being the diameter of the sun pinion or drive gear, and R the diameter of the outer sleeve or gear ring. The stroke is determined by the eccentric throw of the eccentric unit.

It might be well to mention at this point that although gears have been used to illustrate the eccentric unit, rolling members may be substituted for all gearing. With this method the planet gear pin is eliminated and the planet gear becomes a rolling wedge between the sun roller and the inner raceway of the outer sleeve.

When using rolling members the mechanical efficiency is appreciably higher than when using gears, as is to be expected, and is also smooth and quiet. To offset these advantages, however, there is a practical limit to the power that can be transmitted due primarily to the fact that all rolling surfaces should be equivalent to those of an antifriction roller bearing.

Rolling eccentric units up to three horsepower have tested satisfactorily. Best results were obtained using SAE 52100 steel, heat treated as follows: Hardening temp., 1500-1535 F; drawing temp., 375-400 F; quenching medium, oil. The temperature of the oil should be not less than 90 F. Hardness is 61-62 Rockwell C.

One other limitation to the use of rolling members is where several eccentric units are required to be driven from one driving shaft to produce different speeds and strokes but still remain synchronized with each other.

Cross section through a typical geared unit is shown at left. The ring gear is clamped in the outer sleeve which could readily be an integral part of a connecting rod. This sleeve may assume any external shape desired.

Torque reaction of the sleeve is counter to that of the eccentric. Therefore, when used in the form of a connecting rod to drive a piston, the torque tends to alleviate piston side thrust and also counteract directional running friction of the eccentric mass.

To support heavy torque loads and thrusts the units may be mounted on journal extensions of the slow speed eccentrics thus relieving the high-speed

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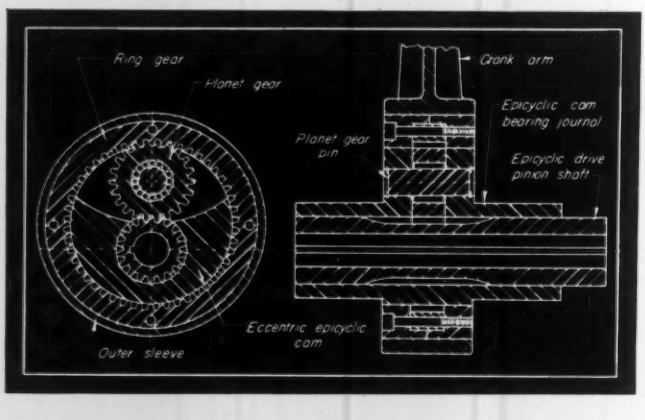
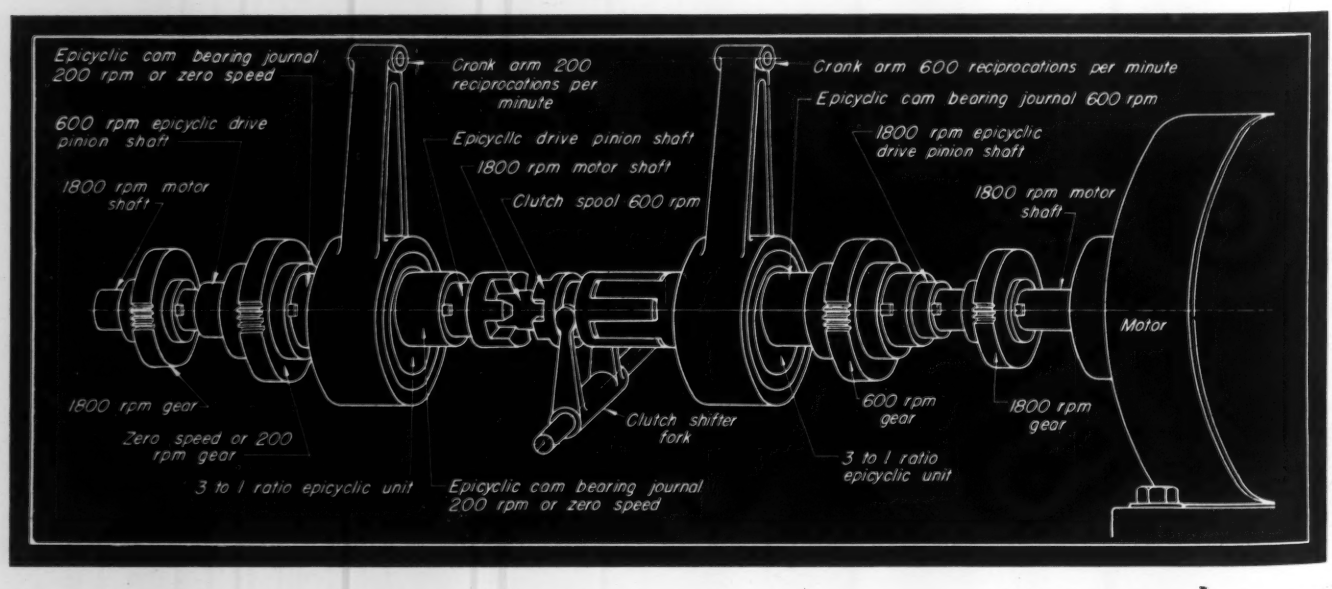
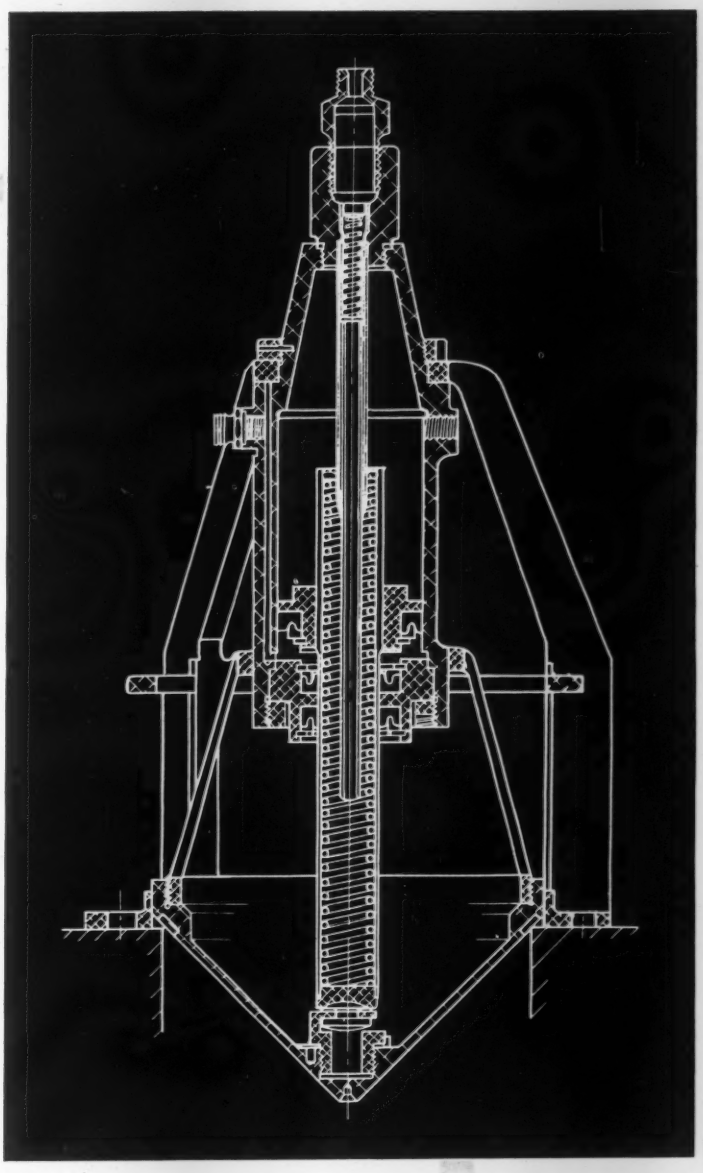



Epicyclic cam bearing journal 200 rpm or zero speed
600 rpm epicyclic drive pinion shaft
1800 rpm motor shaft
Crank arm 200 reciprocations per minute
Epicyclic drive pinion shaft
1800 rpm motor shaft
Clutch spool 600 rpm
Crank arm 600 reciprocations per minute
Epicyclic cam bearing journal 600 rpm
1800 rpm epicyclic drive pinion shaft
1800 rpm motor shaft
Motor
1800 rpm gear
Zero speed or 200 rpm gear
3 to 1 ratio epicyclic unit
Epicyclic cam bearing journal 200 rpm or zero speed
Clutch shifter fork
600 rpm gear
3 to 1 ratio epicyclic unit
1800 rpm gear



Controlling Pneumatic Cylinders

Analysis of pneumatic cylinder design, outlining the various factors which must be considered to obtain the desired degree of precision control

By James L. Dooley
Consulting Mechanical Engineer
Venice, Calif.

ALTHOUGH linear-motion cylinders using fluid pressures to obtain force and do work take many forms, for the purposes of this article they will be considered to be round, tubular housings of definite lengths, with end pieces suitable for carrying the pressures involved. Since most air cylinders are designed specifically for the job they are to do, only gen-

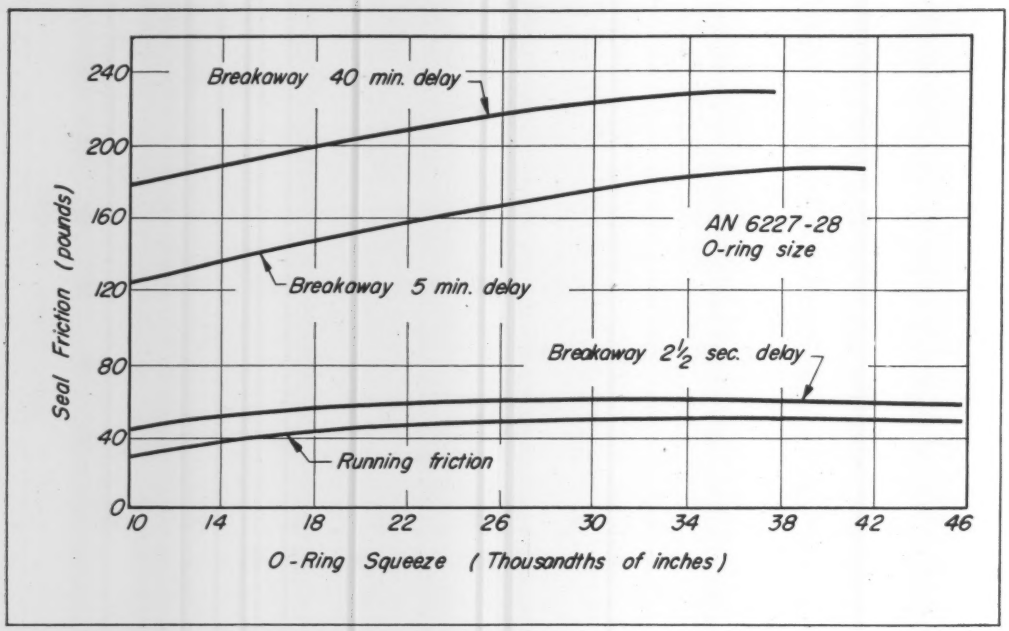


Fig. 1—Left—Characteristic friction curves for different standing times. Piston speed, 300 ipm; lubricant, AN-VVO-366-B; room temperature; surface finish, approximately 3.0 microinches; pressure used, 1500 psi

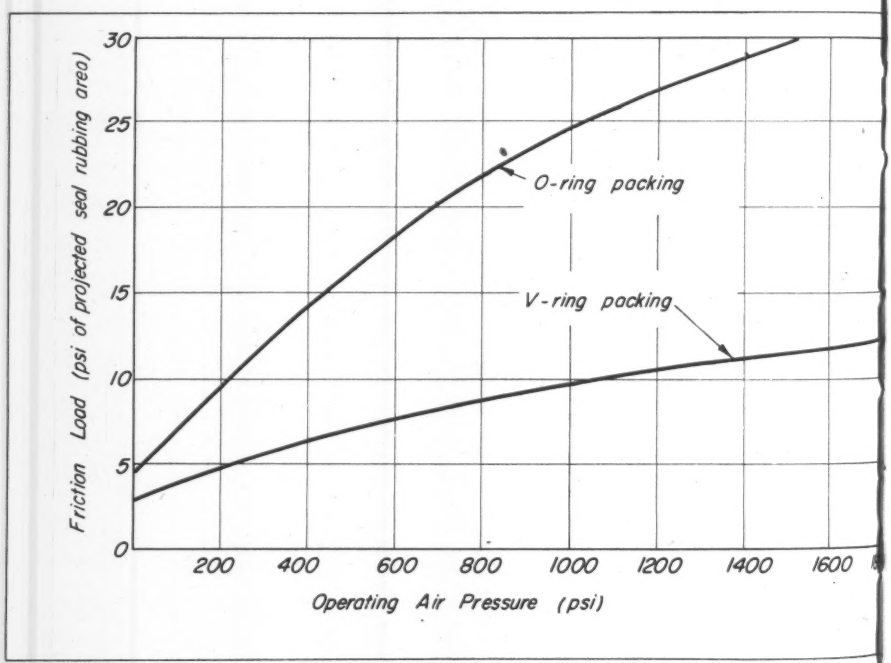


Fig. 2 — Below — Approximate friction load imposed by lubricated O and V-ring seals. Projected area of V-ring is several times that of the O-ring

eral cases and typical examples will be dealt with in this presentation. The methods of mounting the cylinder housing and piston rod alone have many variations, some of which are readily available commercially as standard models, but inasmuch as the methods of mounting vary so much with each application, they will not be considered in detail.

Primary factors influencing pneumatic cylinder design may be itemized as: (a) The forces required at the piston rod throughout the stroke; (b) the air pressures involved (this may be any gas so long as temperature, corrosiveness and toxic effects are considered); (c) length of stroke required; (d) shock loads and side loads; (e) type of service, life, lubrication, exposure; (f) the permissible weight; (g) the number of units required; (h) the permissible cost; (i) adaptability to a variety of applications; (j) the character of load-stroke curve; (k) operating time; and (l) the control required.

AVAILABLE FORCE: The force available from a pneumatic cylinder for external work at any instant during the piston stroke depends upon a variety of factors, each or which should be fully considered.

Pressure Differential: The differential air pressure available within the cylinder under both static and flowing conditions must be considered. Effective inlet air pressure is the main line pressure when standing, but all friction losses and valve pressure drops must be considered under flowing conditions to ascertain the effective inlet pressure when the piston is moving. Tables for estimating these pressure drops at any particular flow are given in handbooks or can be obtained from manufacturers. The back pressure on the piston is generally atmospheric when standing, with additions for all pressure losses incurred in forcing the air out when the piston is moving. Unless a restriction is deliberately added or the piston movement is very fast, the back pressure friction and valve losses can be neglected.

Piston Area: The effective piston area is the area of the cylinder bore minus the area of the piston rod or rods where necessary. Cylinders having a rod at one end only naturally will have a "pull" load different from the "push" load.

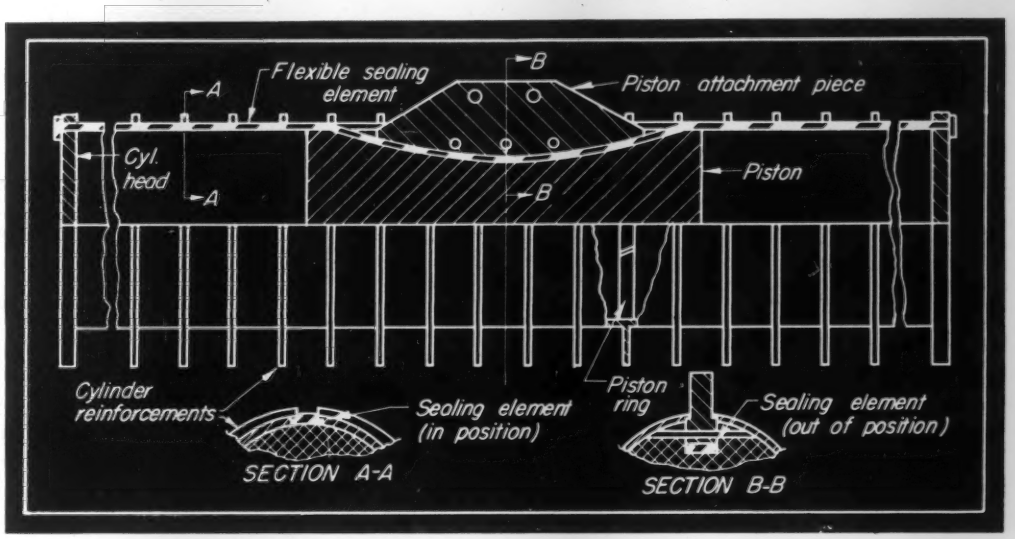
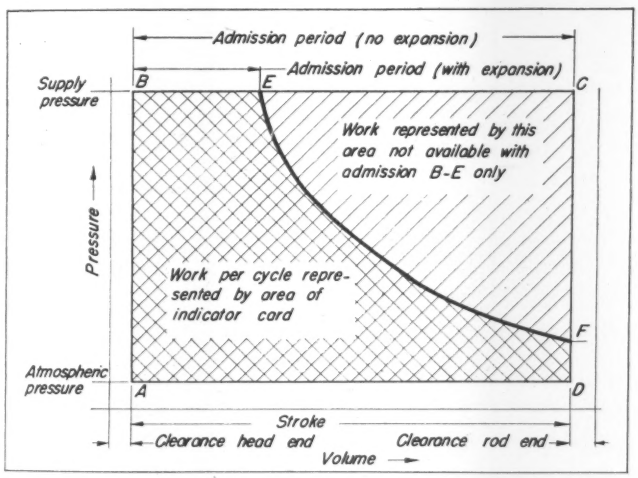


Fig. 3—Right—Cylinder and piston design for long stroke where no piston rod can be used. Piston attachment passes through a slot in the cylinder wall which is sealed against pressure with a flexible sealing element

Fig. 4—Below—Ideal indicator card diagram for an air cylinder



Friction Losses: Sealing elements as well as side loads on the piston and piston rod create frictional losses. Breakaway friction is always higher than running friction and depends on the standing time, load, lubricant and temperature. Sealing friction is a function of the type of packing, the particular installation clearances, the operating pressure, the surface finish, the lubricants used, the operating temperature, the speed of the piston, the character and age of the packing and probably several other things including the installation technique. Although it is difficult to estimate this friction accurately, it is not generally serious as it is small compared to the other forces. Means of roughly estimating O and V-ring seal friction are given in *Figs.* 1 and 2. As for leather cup type packing, little data has been accumulated but friction seems to vary from 3 to 20 pounds running drag per lineal inch when lubricated and at normal shop pressures of 50 to 100 psi.

Effects of Leakage: The usual air cylinder application has seals which make this factor negligible. In cylinders that seal at the end of the stroke, but do not incorporate seals on the piston for operation during the stroke, the effects of leakage past the piston may have to be taken into account when estimating the effective pressure. This also is true of the type of cylinder shown in *Fig.* 3, as the leakage along the flexible sealing element is high.

Piston and Piston Rod Acceleration Forces: Proper acceleration is only of importance when high piston speed is involved and where it is required to know the force available at the piston rod accurately throughout the stroke. Even then these inertia forces are generally small compared to the inertia forces of the load. When required it can be obtained from the formula $F = wa/g$, where F = force required at any instant (lb), w = weight of piston and rod (lb), g = gravitational constant (32.2 ft per sec per sec), and a = acceleration at same instant during the stroke (ft per sec per sec).

There are several other factors which are of interest in air cylinder design that are related to the forces available and the operating time. The first is head clearance at the end of the stroke. In a hydraulic cylinder, extra head clearance is of little importance as the fluid filling this space is relatively incompressible, but in air cylinders this space must be filled with compressed air each time the pressure becomes effective on the piston. This is not only wasteful of the air but slows the operating time and makes accurate control of the piston more difficult.

Cold Walls Increase Operating Time

The second is cylinder temperature. If the walls of the cylinder are very cold, the operating time is increased not only because of the increased drag, but because the air weight required is greater due to gas shrinkage. Unfortunately cylinders utilizing the expansion of the air, as shown in curve *ABEFDA*, *Fig.* 4, are cooled by isentropic expansions of the air itself. This is of little consequence, however, unless the cylinder is operated continuously. The average air cylinder installation does not warrant the complications in valving necessary to cut off the admission before the end of the stroke to save air even though the load-stroke relationship may permit it.

In designing pneumatic cylinders for special appli-

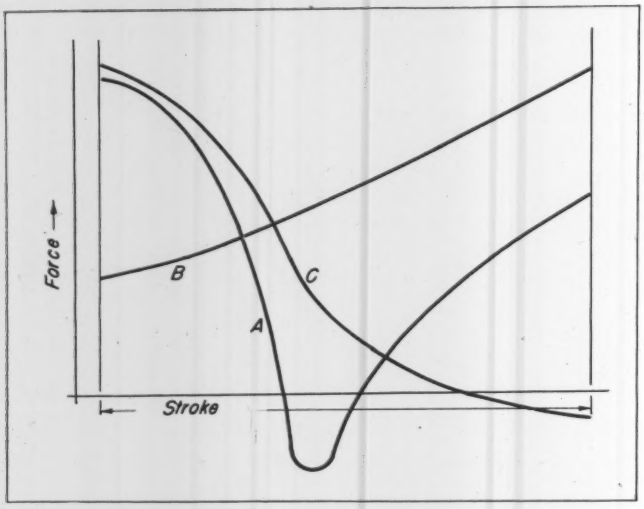


Fig. 5—Characteristic cylinder load-stroke curves

cations, consideration should also be given to the following points:

1. Bore and stroke required to handle the load with the minimum pressure available
2. Type of snubbing or the control required
3. Type of cylinder and piston rod connections
4. Packing best suited to the application
5. Life expectancy
6. Most economical construction
7. Stress in parts
8. Bearing length when extended
9. Protection of piston rod when extended and cleansing of rod before re-entering packing during retraction
10. Nature of gas used in cylinder. Nitrogen gas at 450 psi, for instance, is used to power the single-acting cylinder of the 1000-gpm alcohol valve used in the German V2 rocket (see cross section on lead page of this article)
11. Ease of maintenance and servicing.

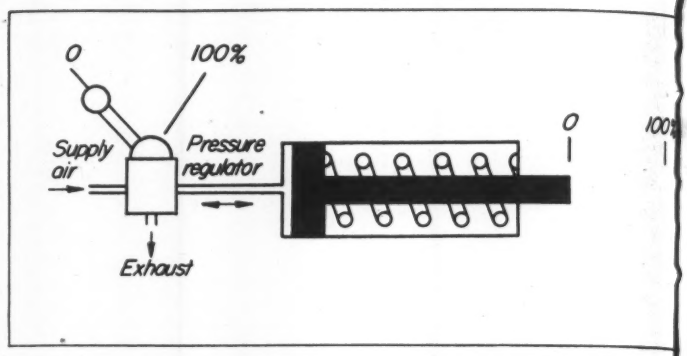


Fig. 6—Schematic arrangement of cylinder and regulator to allow piston to stop at intermediate positions

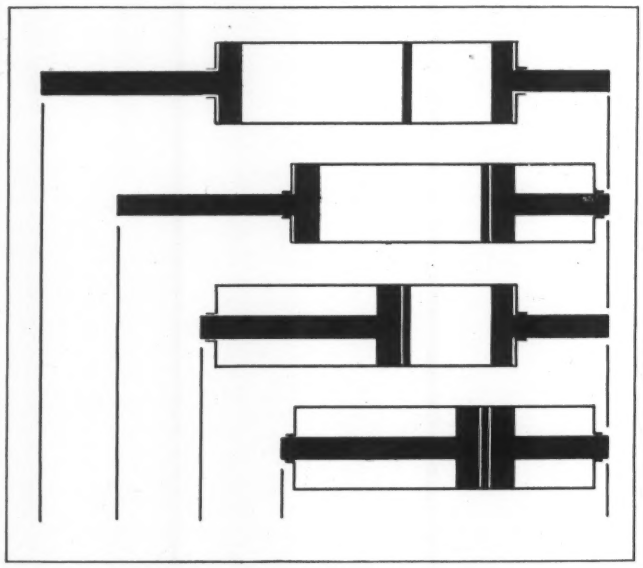
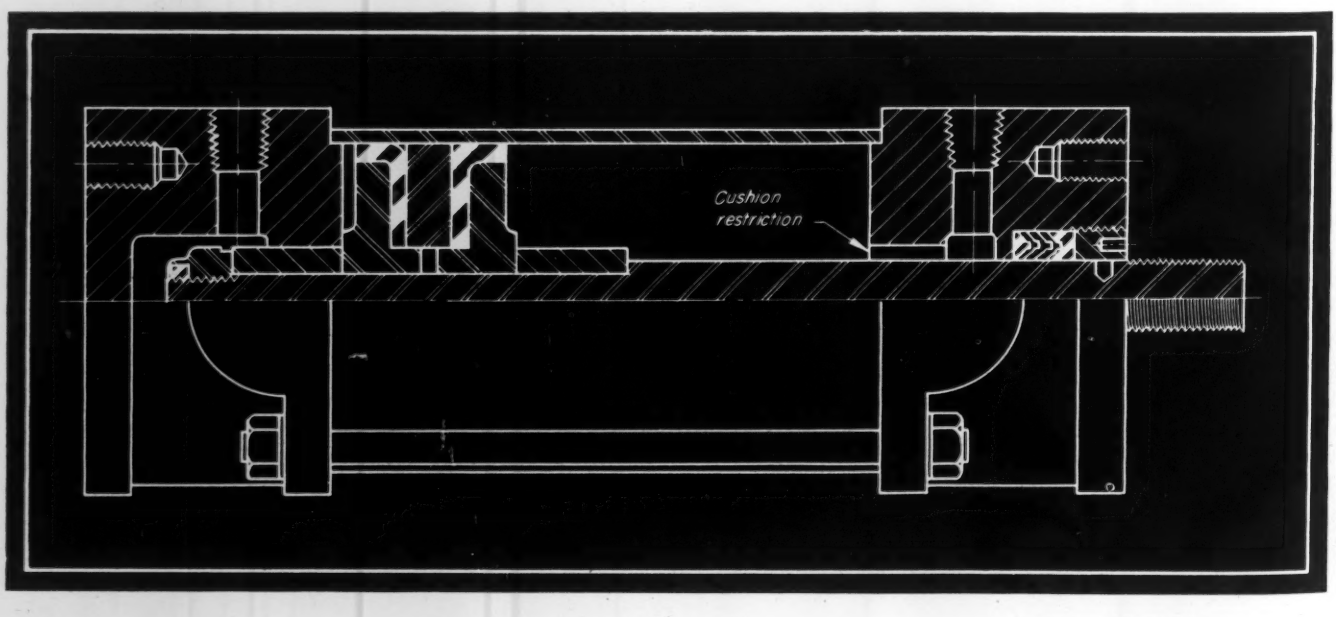
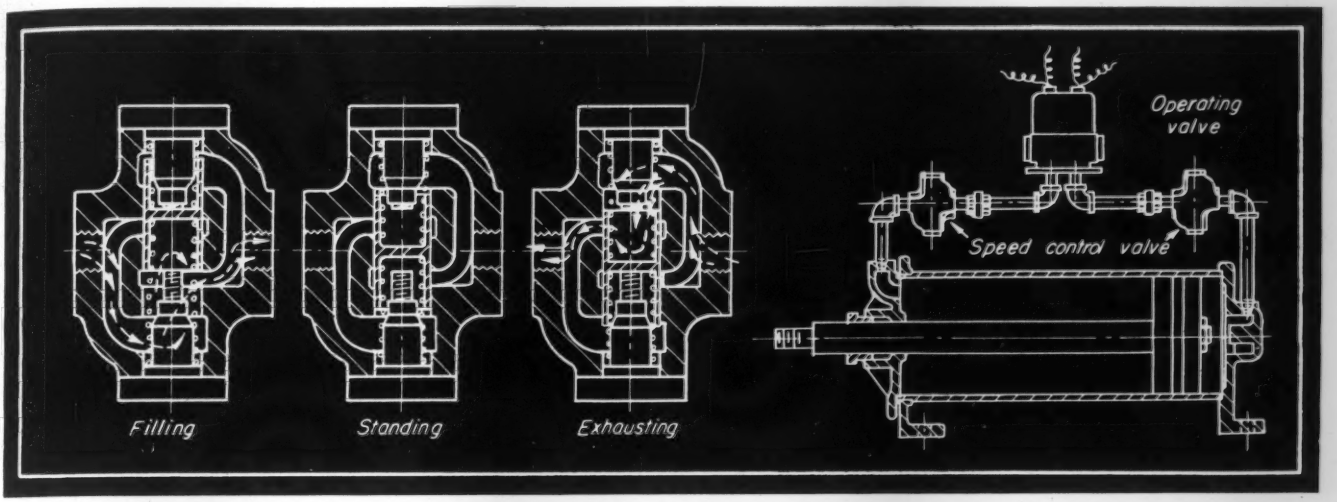


Fig. 7—Above—By arranging two cylinders of different strokes back-to-back with flexible connections, four definite positions are available. Two four-way valves are used

Fig. 8—Below—Typical heavy-duty industrial cylinder for moderate pressures. Cushioning occurs at end of stroke

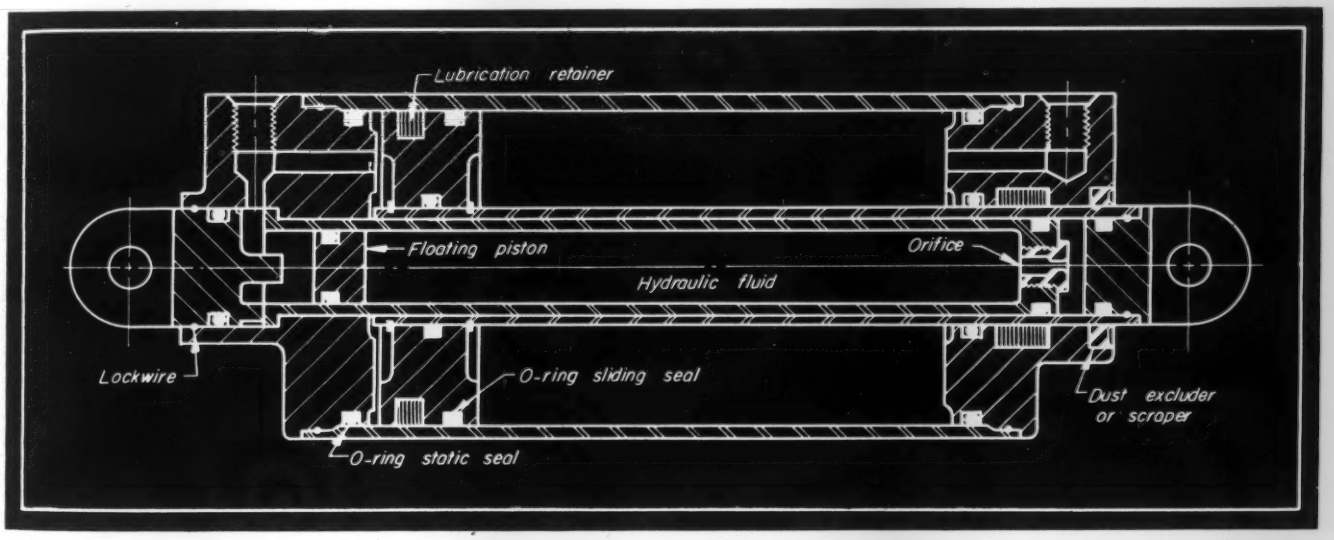




Drawing, courtesy Hanna Engineering Works

Fig. 9—Above—Installation and operation of control valve for regulating piston speed in either direction

Fig. 10—Below—Pneumatic cylinder of lightweight design with integral hydraulic damping

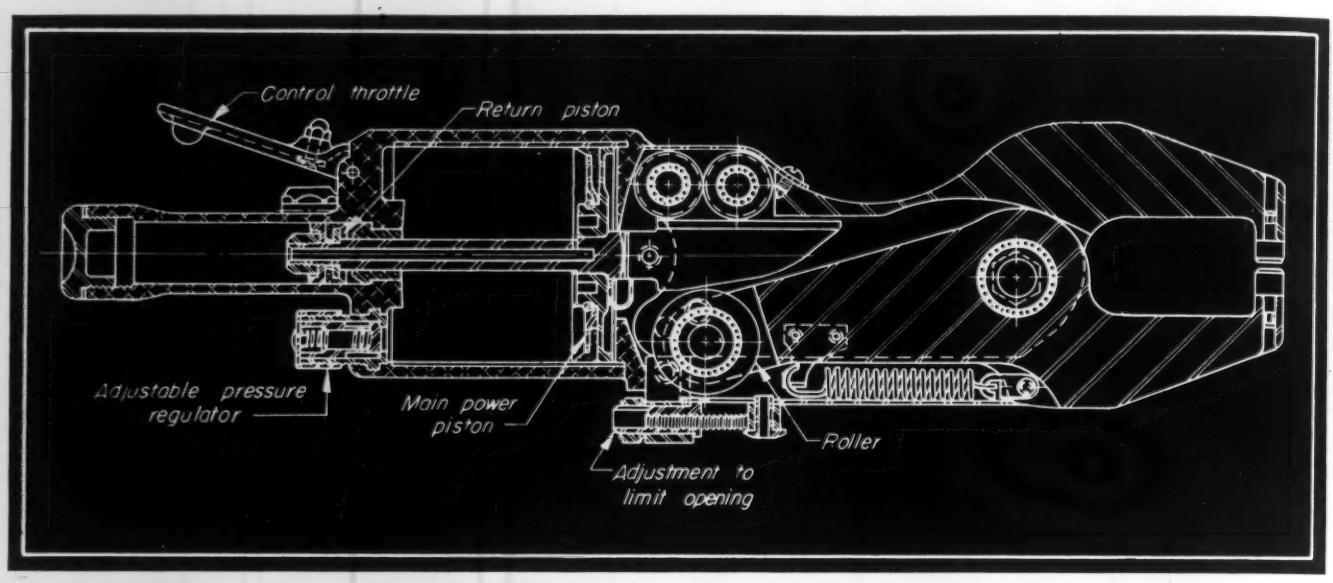


CONTROL OF PNEUMATIC CYLINDERS: In general, the control of air cylinders must be given more consideration than hydraulic cylinders with many jobs because the operating medium is highly compressible. Actually hydraulic fluid is also considered compressible for many high-speed, high-pressure, dynamic applications, but this compressibility in either gases or fluids often may be advantageous when properly used as exemplified by the liquid springs being used in aircraft service. The compressibility of air puts it at a disadvantage only when very smooth starts or when accurate but varying intermediate positions are required with an odd load-stroke curve such as *A* in *Fig.* 5. When the load-stroke curve is continuously rising as in *Fig.* 5, curve *B*, the air cylinder can be arranged to take an intermediate position by controlling the differential pressure across the piston as in *Fig.* 6. This arrangement is being used on remote controls for industrial (and recently aircraft) engines as well as in every cylinder type industrial hoist. This method of positioning either air or hydraulic cylinders may not be satisfactory, however, on a varying or fluttering type load.

If definite intermediate positions are required, they can be obtained with multiple air cylinders as shown in *Fig.* 7. Here two cylinders back-to-back give four definite positions. This same result may be obtained several other ways depending upon the forces acting and the nature of the requirements. Automatic pneumatic stops, detents, or a linear equivalent of a Geneva movement may be considered.

Desired Control Features Easily Obtained

The great majority of applications require only snubbing or cushioning at the end of the stroke to prevent banging and some type of rate control. By controlling the admission rate and by installing an end cushion as shown in *Fig.* 8, both of these features



Drawing, courtesy Chicago Pneumatic Tool Co.

Fig. 11—Above—Interesting application of single acting air cylinders in lightweight, powerful riveter. Large piston provides power stroke and small one in handle, return

can be obtained with little difficulty.

To control the speed of movement throughout the stroke, the line pressure can be throttled and the exhaust line restricted to maintain not only a definite differential pressure across the piston but to maintain a high air density in the cylinder to effect good damping. Restrictors are usually installed in the air lines to the cylinder to do this as shown in *Fig.* 9. Note that the rate of fill and drain can be separately adjusted.

Normally one end of the cylinder or the other is filled with compressed air which is desirable when a maximum amount of restricting is desired. In special cases where equipment may be damaged if proper restriction is not present, it is necessary to design the control equipment to insure this cushion air change. With these two controlling factors, it is possible to effect very good control under the most adverse loading, for example, one where there is a high initial load to overcome that falls to a low negative value before the end of the stroke, *Fig.* 5C.

Another variation of the air exhaust restrictor is

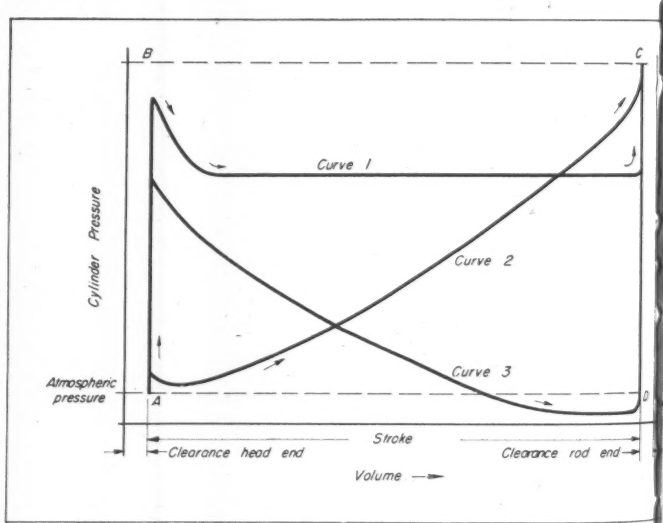


Fig. 12—Above—Actual indicator cards for an air cylinder under various loading conditions

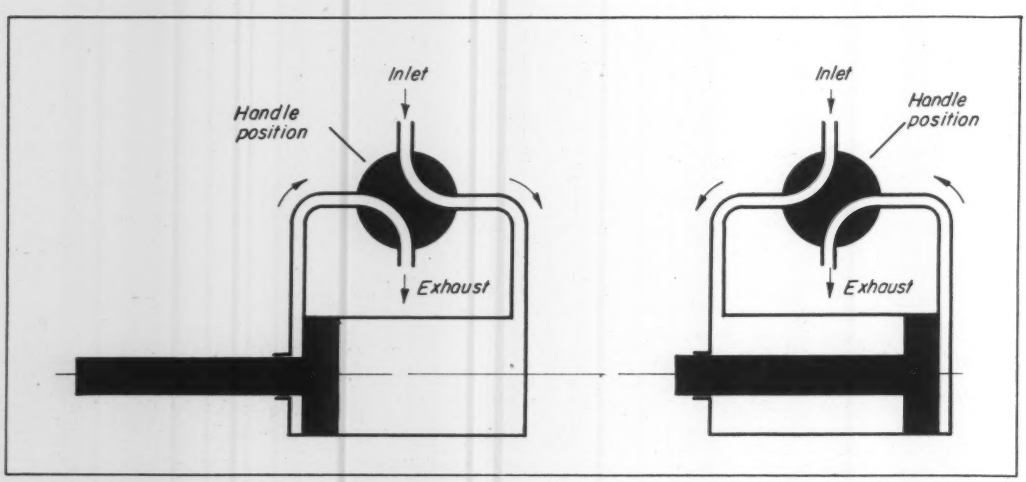


Fig. 13 — Left — Schematic diagram showing requirements of typical four-way valve to move piston in either direction

to permit free exit of air over the greater portion of the stroke, after which the main exhaust port is covered with the piston so that the remaining portion of the stroke forces the exhaust air through a restriction for cushioning.

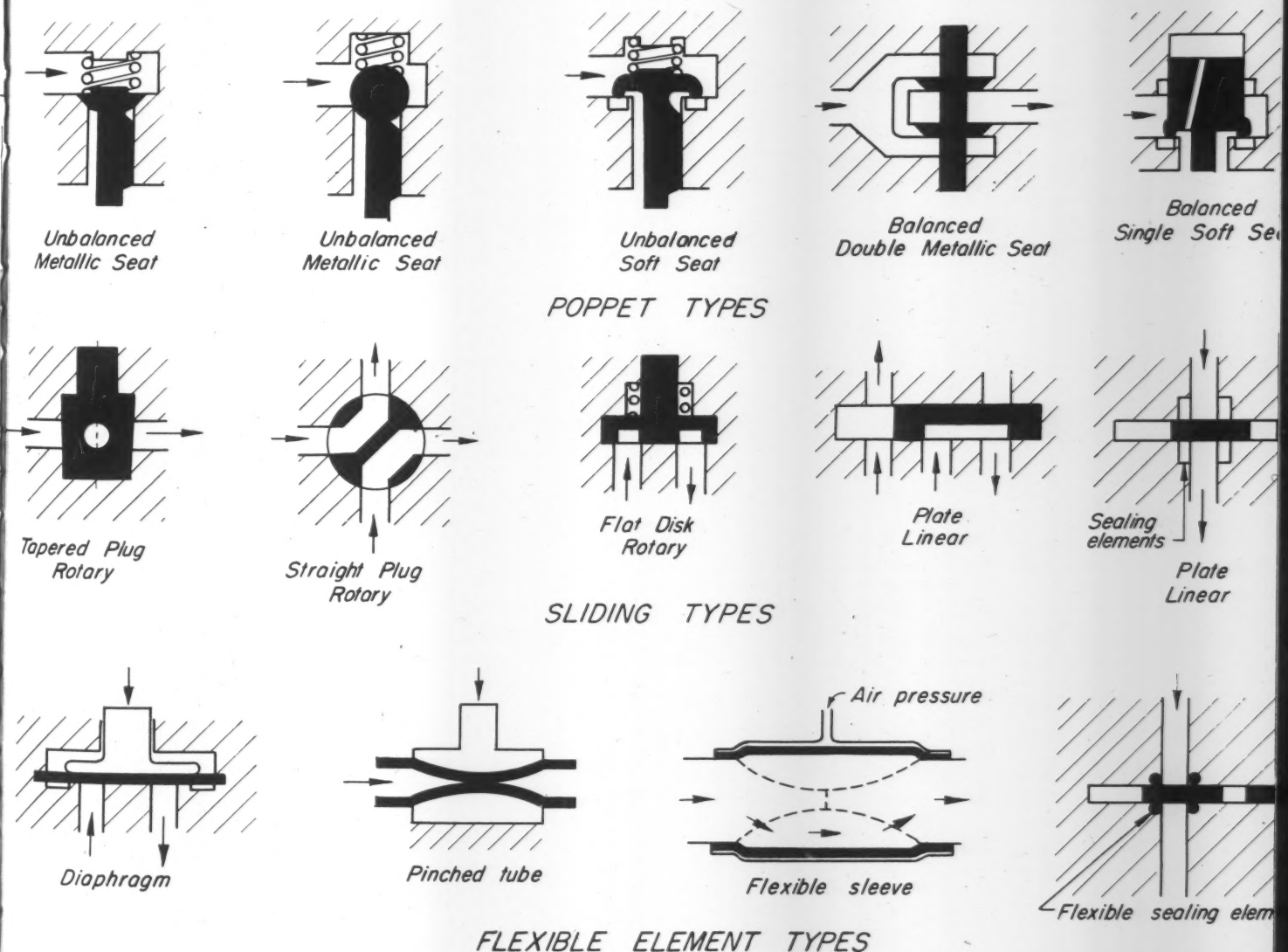
Where definite speed control is required with all types of load curves, as in a pneumatic cylinder automotive lift, it is generally advantageous to use fluid to control the piston speed. To do this the air pressure may be applied to the hydraulic fluid in an accumulator and a hydraulic cylinder used, or hydraulic control can be built into the cylinder. A separate hydraulic or mechanical snubber can be used or built into the air cylinder as shown in *Fig.* 10. One might wonder why a hydraulic system is not used entirely if it is necessary to resort to hydraulic cylinders for control, but the air system is lighter, has less transmission loss, and may be less expensive than to install the requisite hydraulic pumps especially where compressed air capacity is already available.

An interesting application of an air cylinder where pressure control rather than speed control is important is shown in *Fig.* 11. This pneumatic rivet squeezer uses two single-acting air cylinders and a wedge type mechanical advantage to apply a high but regulated load to the rivet. The small piston in the handle has line pressure applied to it at all times for the return stroke while a three-way valve applies air pressure to the main cylinder for operation and drains it for release.

Several typical indicator cards for an air cylinder under different loading conditions are shown in *Fig.* 12. The required pressure for movement is generally somewhat high initially for breakaway and falls to almost that required for the load alone during the major portion of the stroke, building up again near the end when the cushion device comes into play.

(*Concluded on Page* 196)

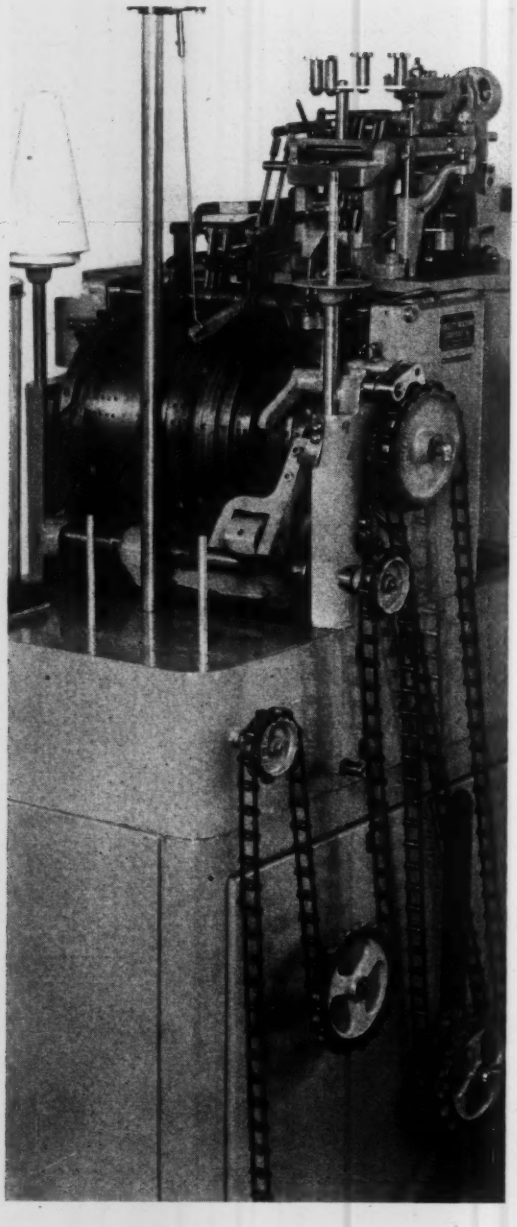
Fig. 14—Below—Schematics showing typical valve types



Unusual Drives . . .

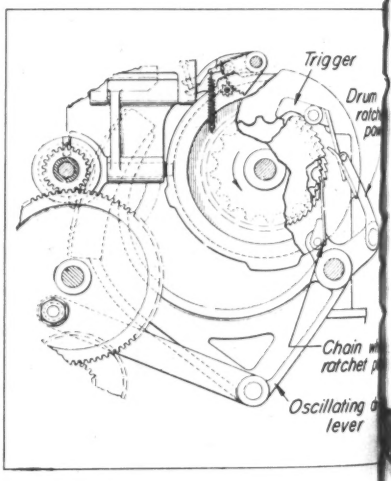
and control elements of a selection of interesting machines indicate the broad possibilities in this field

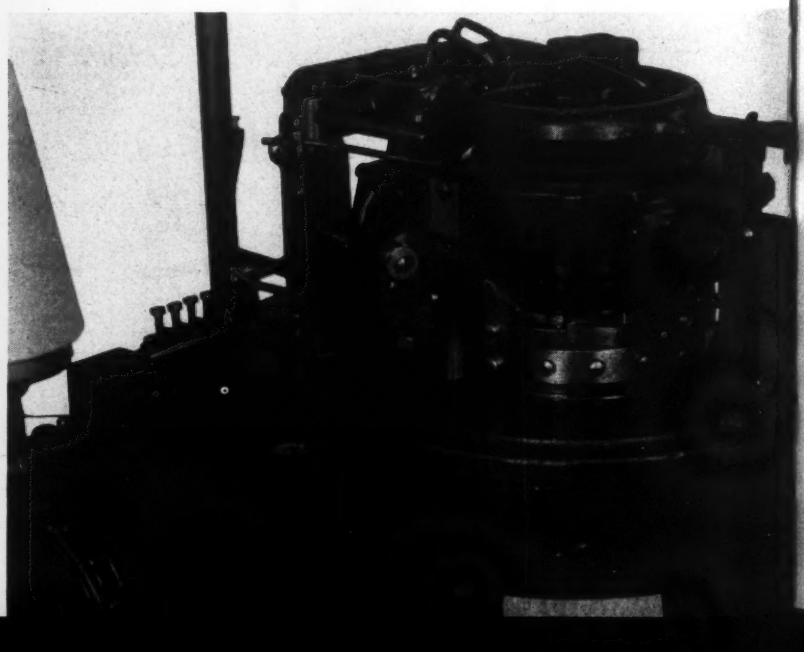
Pattern Chain Drive



An endless pattern chain on the Fidelity 400 circular knitting machine, left, controls the ratcheted advance of the pattern drum from which all the functions of the machine are governed. Yarn changes, number of courses in each portion of a stocking, knitting speed changes, etc., are all initiated and regulated by the master cams of the pattern drum in accordance with the arrangement of the "operational" links in the chain. These links are spaced to provide the desired cycle.

Continuously actuated by an oscillating lever, the chain ratchet pawl moves the chain while the pattern-drum ratchet pawl is on its backward stroke. When a cam on one of the operational links in the chain raises the control trigger, the drum pawl is allowed to engage its ratchet wheel on the forward stroke. The drum is thus advanced while the chain is idle. As the drum pawl retracts, the chain is again advanced, four strokes of the





ratchet being necessary to move one link length. Number of advancing strokes taken by the drum ratchet thus depends upon the shape or length of the cam on the chain link or links and as the trigger drops off the chain cam, the pattern drum pawl is again rendered inoperative until the next operational link reaches the trigger.

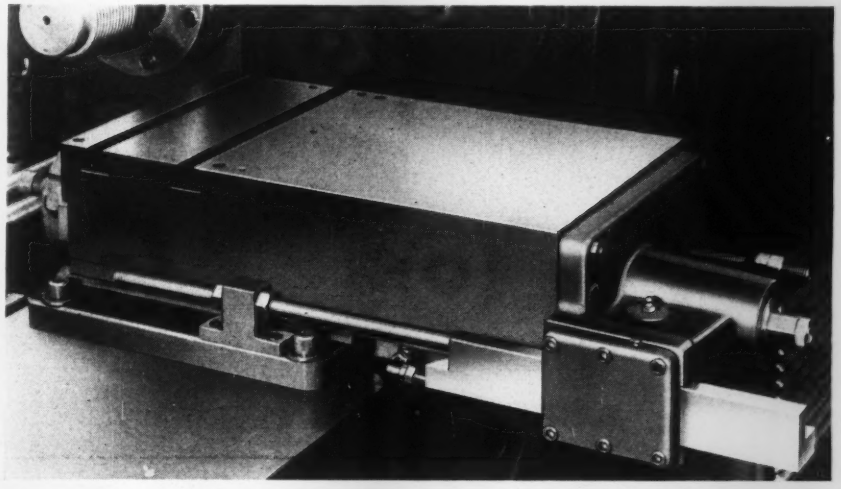
A speed-change cam at the far left of the pattern drum controls, by means of a rheostat in an electronic motor circuit, the speed of the driving motor in direct relation to the various mechanical changes effected in the knitting cycle.

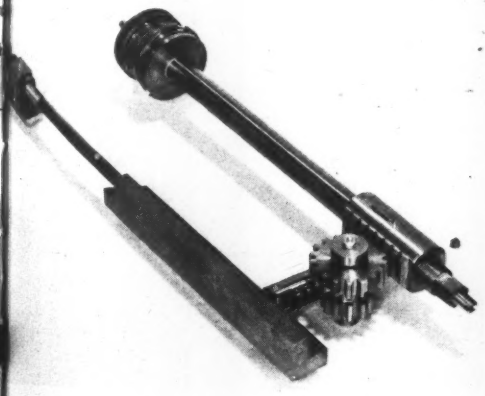
Mechanical Escapement

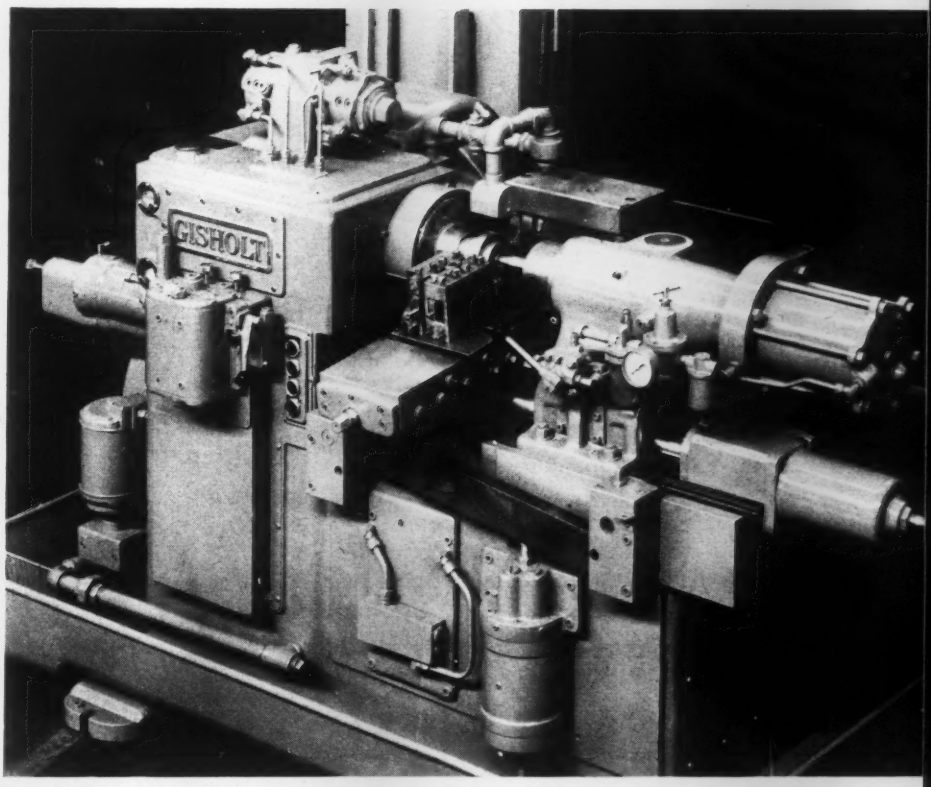
To obtain a feed slowup or stop for deep forming or grooving with the rear carriage, a unique mechanical escapement is provided in the Gisholt No. 12 automatic lathe, bottom right. Simple but effective, the device shown below comprises a cluster gear, the larger gear of which meshes with rack teeth on the nut of the piston shaft and the smaller gear with teeth of a roller-ended rack. As the piston advances the rear slide, right above, the roller of the small circular rack travels along a flat, fixed bar. When the roller reaches the drop-off angle of the fixed bar, piston travel remains the same but carriage travel is reduced according to the angle of taper on the bar owing to action of the cluster gear escapement.

All tool travel—rapid approach, feed during a cut, tool fadeout, and rapid retraction to rest position at the end of the cycle—is accomplished by hydraulic actuation. One hydraulic system furnishes both drive and control; limit switches, cams, solenoids, mechanical trips, etc., are dispensed with. Carriage control is effected entirely by pressure variation within the hydraulic lines.

In operation, the vertical piston on the front of the machine elevates the guide bar, advancing the tools of the front carriage transversely to cutting position whereupon the piston at the headstock end moves the carriage longi-



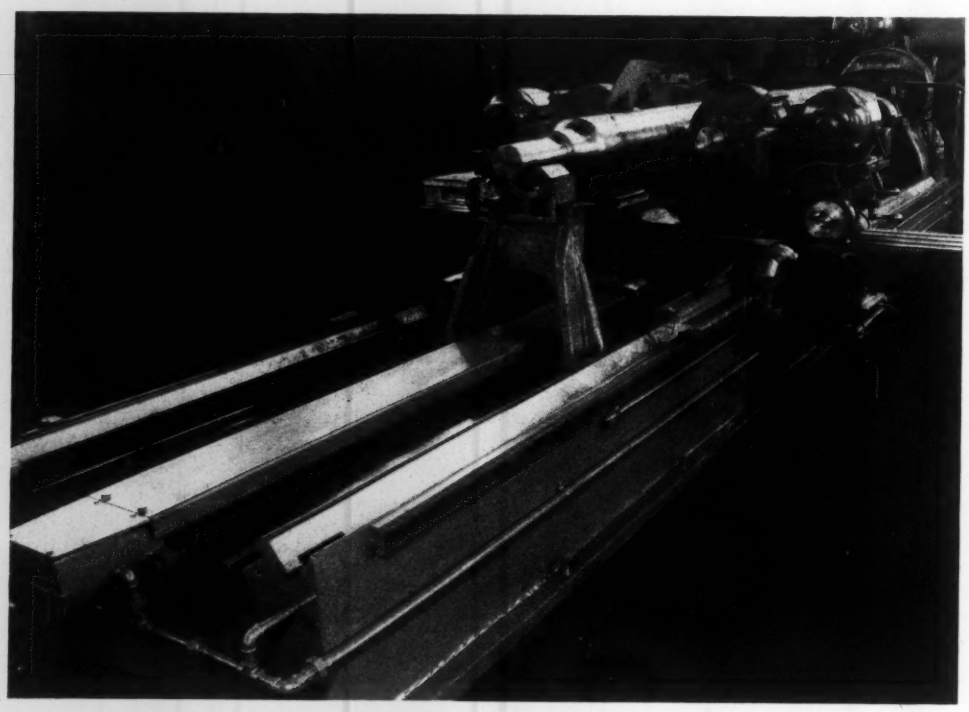




tudinally to complete the turning. On completion of the cut the tools retract both cross and lengthways by controlled hydraulic relief. Simultaneously with the front slide the rear piston advances the rear tools rapidly to the work and feeds them through the cut, rapidly retracting the slide along with the front carriage. As retraction commences a piston on the headstock disengages the clutch and applies the brake.

Automatic In-Feed

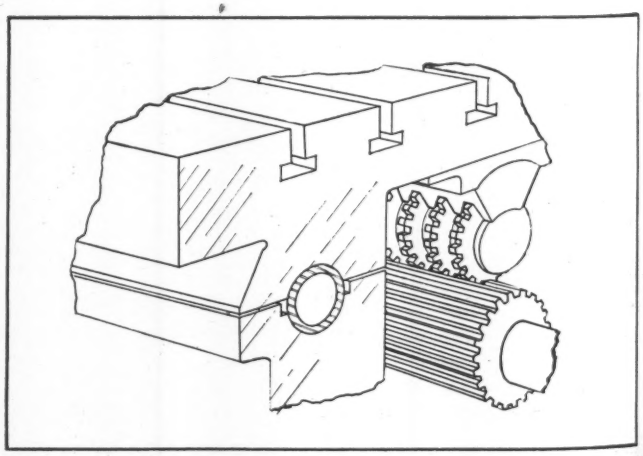
On the Lobdell roll grinder, below, an automatic in-feed device feeds the grinding wheels toward the work gradually as they traverse along the face, maintaining a constant pressure between the wheels and the work as the grinding surface wears away. Adjustable to provide any desired rate of in-feed from extremely slow to as fast as the work and wheel necessitate, the mechanism, which can be seen directly below the wheel drive motors, is used in conjunction with the regular roll crowning unit. A crank rod, actuated by the gear which engages the longitudinal crowning drive rack, oscillates the short rack on the wheel head saddle. Stroke of the crank rod is adjustable and the oscillating racks are connected across the saddle by a tie rod. Oscillation of the racks rotates the escapement wheels which they engage and effects an automatic elevation of each of the opposed hinged wheel heads, feeding the wheels toward the work. By maintaining a constant and uniform grinding pressure, wheel wear is reduced to a minimum and the grinding operation is improved.



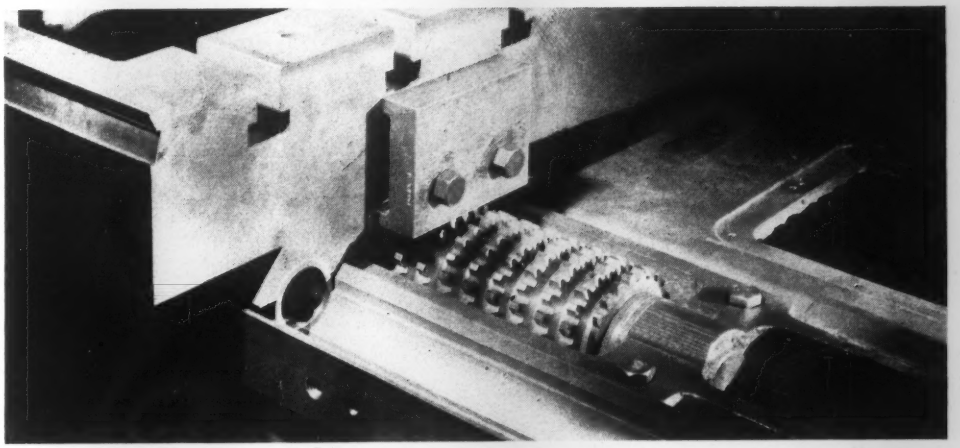
Combination Worm

Drives for reciprocating machine beds of the rack and pinion or rack and worm type sometimes present very difficult design problems. An unusually interesting refinement of the worm and rack type of drive featured on Ingersoll planer type machines is shown at the right and top next page. Worm and rack action is conventional but spur gear teeth are cut across the worm threads so that the combined worm and spur gear may be driven direct by a spur gear located in the bed of the machine.

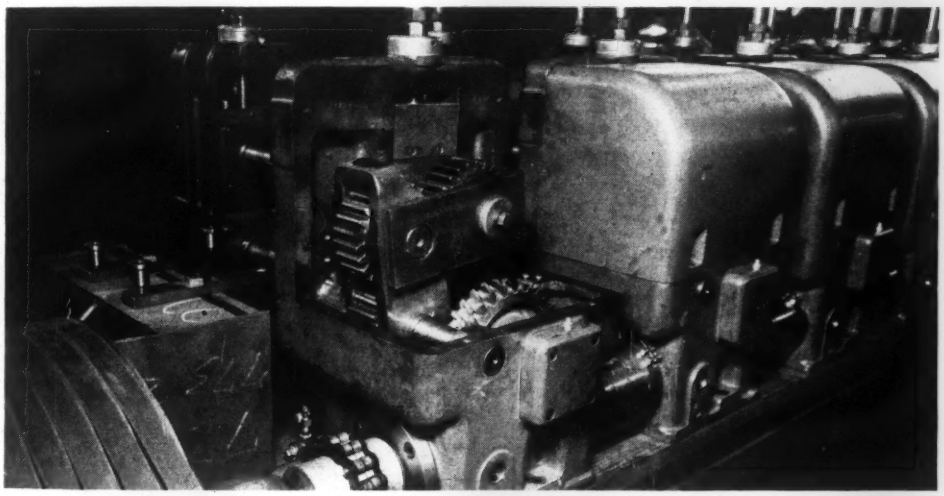
Besides simplifying the drive design and reducing space requirements, this arrangement makes pos-



sible a larger, more rigid drive shaft having considerably less torsional deflection and, in addition, provides maximum tooth contact area reducing wear to a minimum. Smooth feed is obtained regardless of the heaviest interrupted cuts and good lubrication is provided by the worm-spur teeth which lift oil from the reservoir to the rack continuously.

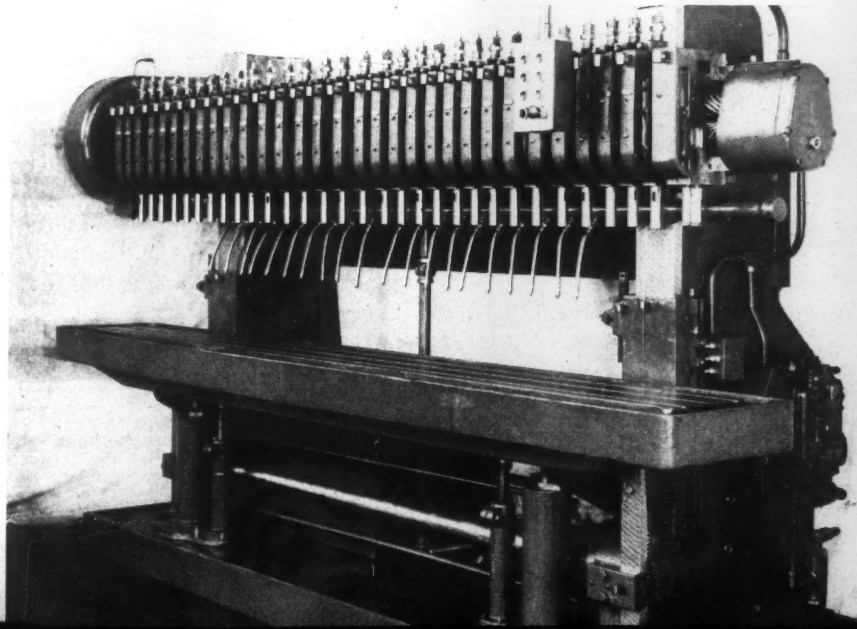


Floating-Center Gear Drive



Micrometric adjustment of the roll spindles of Yoder cold-roll forming mills is accomplished without incurring any serious drive or operational problems. By employing a double floating-center type of drive arrangement, as shown at the left, the upper spindle driving gears of the individual roll stands remain in proper mesh regardless of the vertical center distance required between the rolls.

Hydraulic Cylinder Equalizer



Two hydraulic cylinders located beneath the work table of the Moline HD-13 driller, left, feed the long, narrow table automatically with rapid traverse, feed and/or jump feed or dwell and rapid reverse. To insure precisely uniform table movement by each of the two cylinders, interconnected equalizing pinions, meshing with a rack at each end of the table mechanically link the travel of the hydraulic table drive pistons.

Sequence Control Uses Endless Belt

By Philip W. Vessey
Engineer
Dienelt & Eisenhardt
Philadelphia

AUTOMATIC sequence controls of the adjustable type are particularly suitable for production machines where changes must be made in the manufactured item without slowing down or stopping the machine. An example of this type of control is a device used to actuate a printing machine in the making of felt-base rugs.

Rugs of this type are printed on a continuous machine through which the felt-base material is fed with an intermittent motion, pauses being occupied by the printing stroke. Each printing block measures approximately 18 by 96 inches. Inasmuch as the design may not be a continuous or allover pattern, printing strokes must be carefully synchronized. Also, the printing heads used for the border must be kept out of operation when the body portion of the rug passes. Conversely, the body blocks must not print when the border section is presented to them.

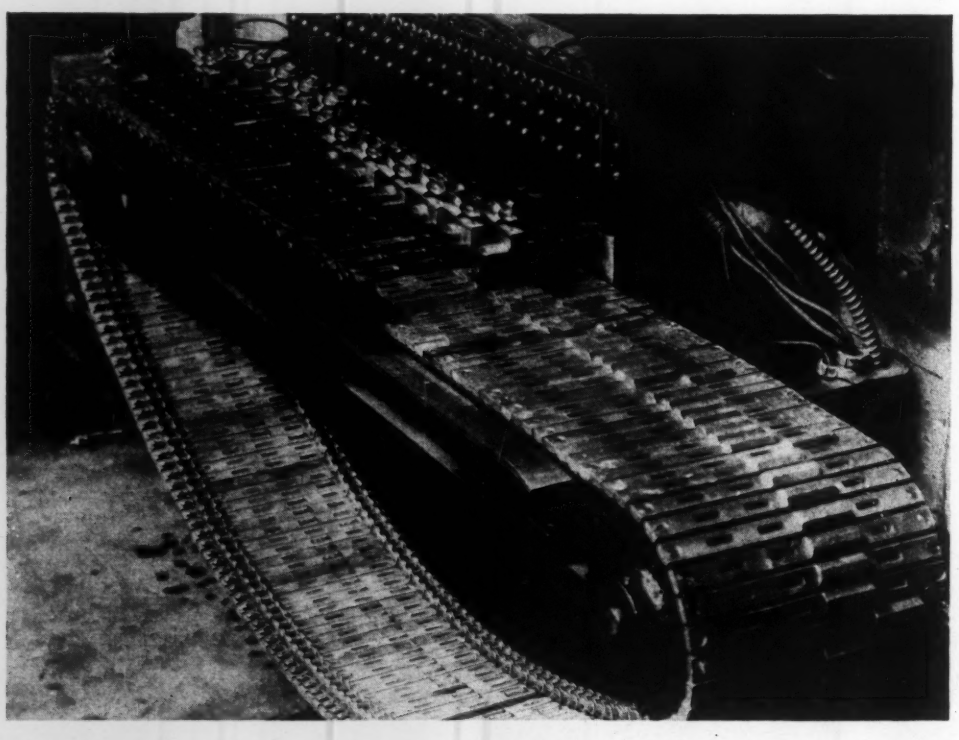
The control device for regulating this printing operation is an endless belt consisting of metal plates fastened at each end to sprocket-driven roller chains. This control assembly is shown in the accompanying illustration. It is actually a moving, miniature representation of the goods being printed, consisting of cams representing the pattern and traveling at a speed proportional to that of the material in the machine. The cams are lugs snapped into slots in the plates of the belt. As each cam comes into contact with a mercury switch, an electromagnetic latch at a corresponding printing head is actuated, a single-color block then being printed.

In the illustration, four rows of cam slots may be observed in the belt plates. In the simplest arrangement, one row might be used for printing the border of the rug and another for the body. The location of the cams are varied to suit the pattern being printed.

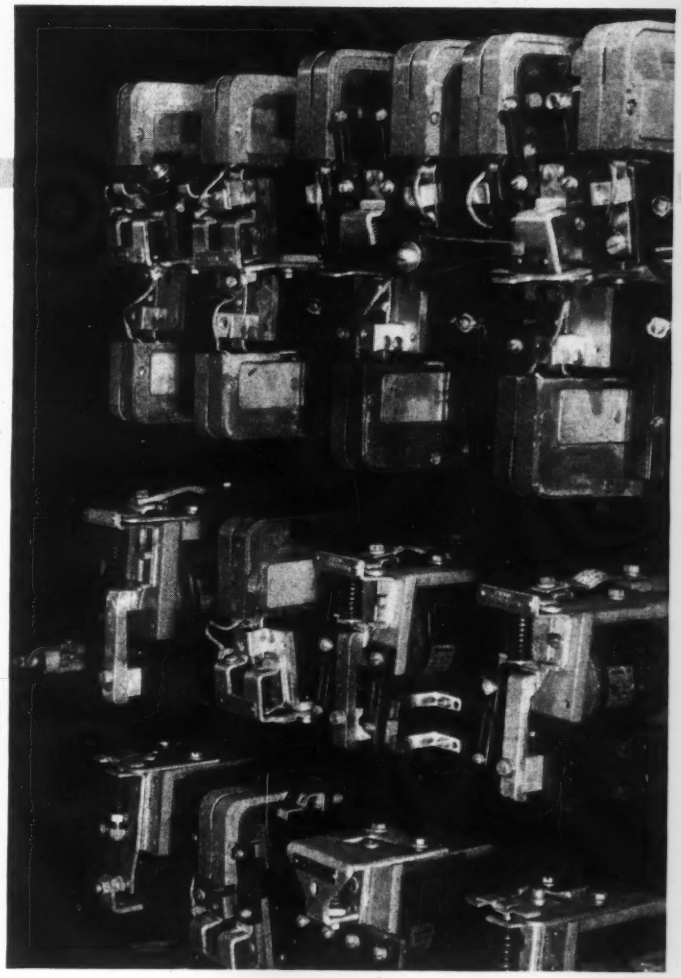
As different lengths of rugs are printed, the arrangement of the cams is made to correspond. A nine foot rug would need six cams while a twelve foot rug would require eight cams.

The operator of the machine has exposed to his view a large portion of the belt, with enough cams to equal the length of several rugs. He therefore has ample time to make any changes in the length or arrangement of the rug pattern before the cams reach the switches. Since the length of the belt is made so that the design will automatically repeat, the operator does not touch the cams except when a poor section of felt comes into position. Then, he changes the cam arrangement so that the bad part will not be printed.

A number of indirect advantages result from the use of this control. As the printing machines are long, the device makes it possible to make all changes at one station. Also, it is not necessary to stop the machine to make changes in the size or arrangement of the printing.



Endless belt control device. The mercury switches seen at the left end of belt are actuated by cams or lugs positioned on the belt plates



Interlocks

. . . protect machines, components and electrical equipment against faulty operation. Basic and typical control circuits involving both electrical and mechanical interlocks are tabulated

By R. B. Immel
Control Engineer
Westinghouse Electric Corp.
East Pittsburgh, Pa.

ELECTRICAL and mechanical interlocking of the component elements of motor starters and control circuits is usually necessary to insure safety for the operator, protection for the motor and driven machine, and a definite sequence in machine operation. The degree or refinement of interlocking necessary is determined only by the specific application. Cost of

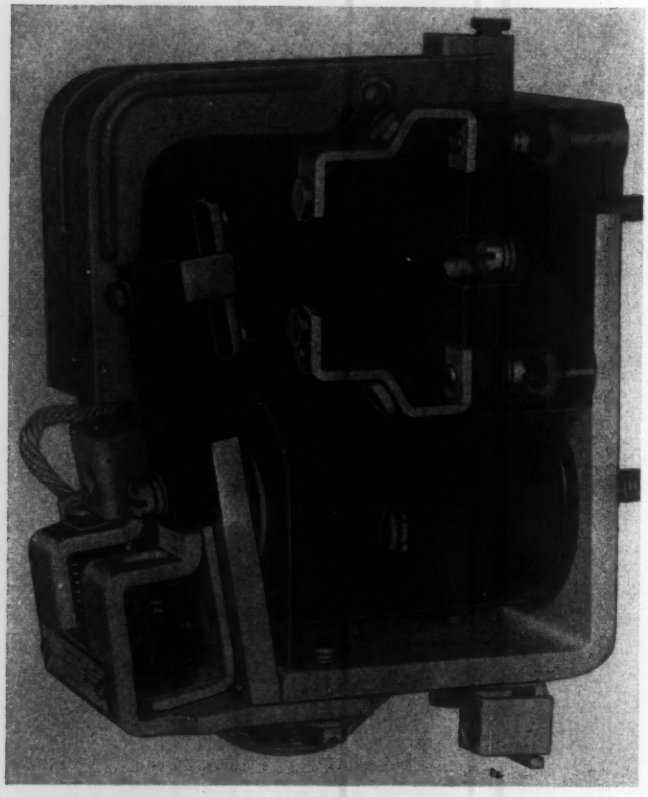
complete interlocking protection is not always justified as the degree of the interlocking and control for a motor drive or electrical system can be determined only by balancing the additional cost of the complete interlocking system against the importance and value of the machine and its product.

Safety for the operator is of paramount imporance and the minimum interlocking protection must always be adequate to achieve this feature.

Interlocking is a term applied to the co-ordination of the internal electrical and mechanical functions of an electrical control circuit, or the governing of the sequence of operation of a group of control circuits and electrical drives. A typical control system, protected by both mechanical and electrical interlocks, is shown in the head illustration. The mechanical interlocking device and the auxiliary contacts for electrical interlocking are on the two contactors in the upper right. Auxiliary contacts on the contactors can be utilized to electrically interlock and control the operation sequence; mechanical interlocking mechanisms can be employed to mechanically insure safety and the required sequence of operation; or relays can be used for interlocking functions.

While the auxiliary contacts on contactors and relays may appear to be insignificant parts, they are extremely essential as even the largest contactor or circuit breaker is impotent without them. *Fig.* 1 shows a typical normally-open, auxiliary-contact assembly mounted on a single-pole, normally-open d-c contactor. When the contactor coil is energized, the

Fig. 1—Typical normally-open auxiliary contact mounted on a single-pole, d-c contactor



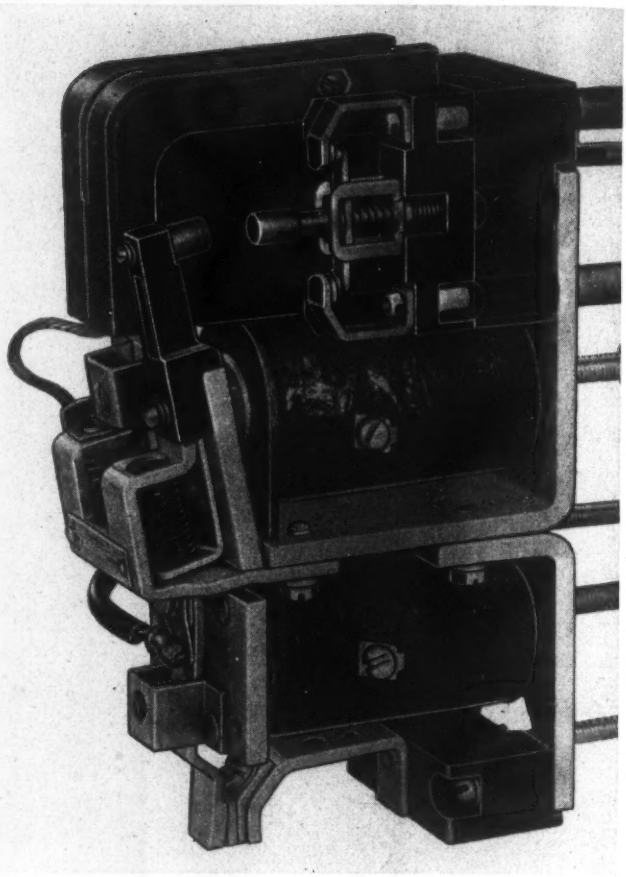


Fig. 2—Delayed type of normally-closed auxiliary contact mounted on a single-pole, double-throw, d-c contactor

contactor armature is attracted to the core pole-face and the moving contact completes the circuit.

A normally-closed, auxiliary-contact assembly may be mounted on the same contactor illustrated in *Fig.* 1 so that when the contactor armature closes, the moving contact interrupts the circuit.

Conventional normally-closed contact assemblies open the circuit almost imediately after the contactor operating coil is energized and the armature starts to move. *Fig.* 2 shows a double-throw, d-c contactor with the upper pole normally open and the lower one normally closed. A "delayed break" type of normally-closed auxiliary contact, which does not open until the contactor armature is practically in the closed or sealed position, is shown mounted on this contactor. The additonal time from the instant that the operating coil is energized until the auxiliary contacts open is sometimes long enough to perform an essential control operation.

The mechanical interlock between two contactors for reversing service is essential for if one contactor should stick closed after its operating coil is de-energized, the closing of the other contactor would short-circuit the power supply.

Use of auxiliary contacts on contactors, mechanical devices, relays, or a combination of these various electrical and mechanical means are illustrated in the following circuit tabulations, showing how

satisfactory interlocking can be achieved. A thorough understanding of these basic interlocking principles will make it possible to provide adequate interlocking for even the most complicated application. For the most satisfactory results, the operation principle for each individual control must be known before any attempt is made to interlock a group of control circuits for a special sequence. While a specific application for a certain circuit in the tabulation is described, the fundamental principle illustrated can be applied to many other control circuits. Except for the use of relays that function only on d-c or a-c energization, in general, the control schemes shown will work equally well for a-c or d-c circuits.

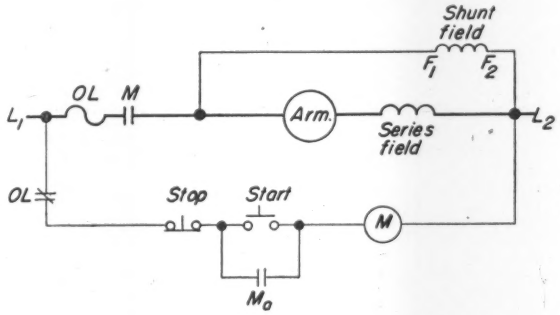
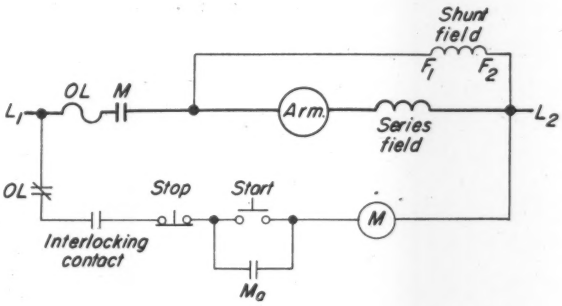
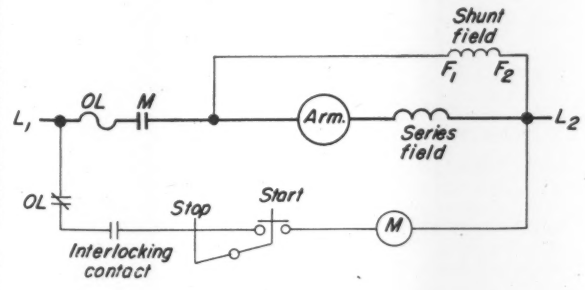
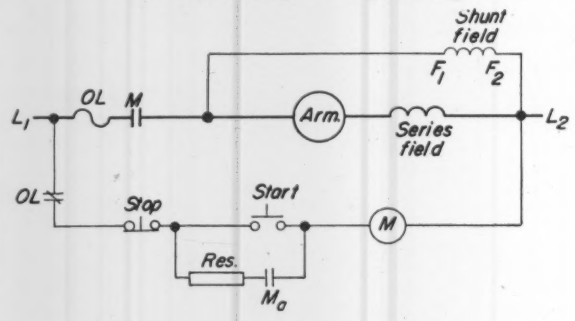
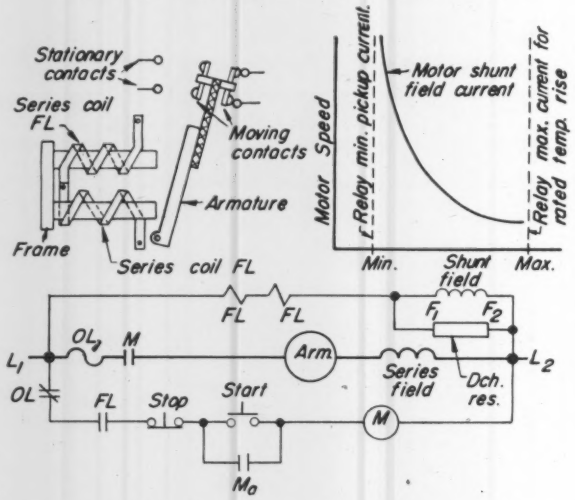
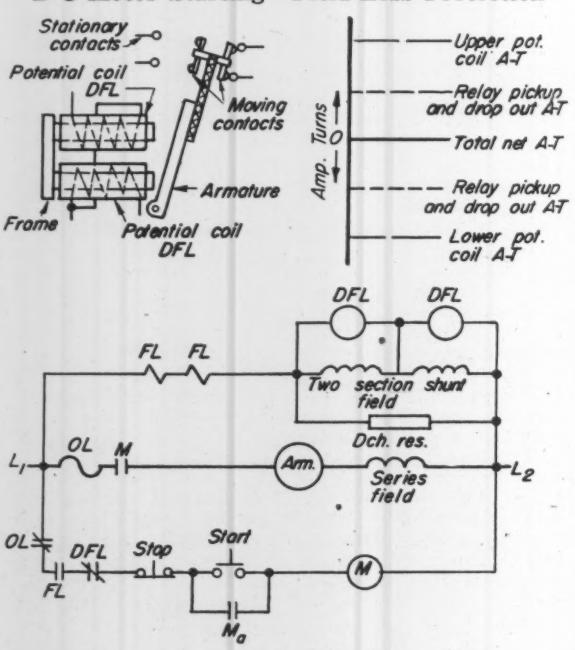
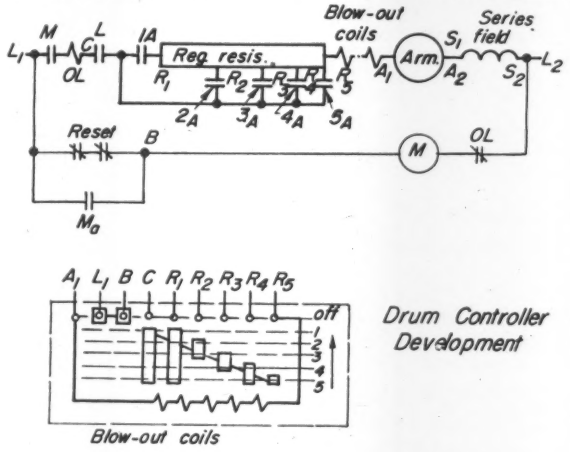
Fig.	Circuit Diagram	Description and Application
3	D-C Motor Starting—Low-Voltage Protection Magnetically Operated 	Low-voltage protection is the term applied to a control scheme which prevents unexpected and unsupervised restarting of a motor which has been stopped because of low voltage or power failure. With this LVP circuit it is always necessary for the operator to push the start button to restart the motor regardless of whether the motor has been stopped by voltage failure, operation of the stop button, or operation of the overload relay. The start and stop button shown are of the momentary-contact or spring-return type. The contactor is designated by *M* and its auxiliary contact which is used for interlocking purposes is represented by M_a. When the start button is operated, a circuit is completed for the energization of the contactor coil *M*. The auxiliary contacts M_a are in parallel with the start button so that a circuit may be maintained to the coil after the start button has been released. If the power fails, the stop button is operated, or the overload relay functions, the circuit to the contactor coil is interrupted and the contactor will open to stop the motor.
4	D-C—Sequence Interlocking of Low-Voltage Protection Motor Starters 	Many machines or processes require that the sequence of operations occur in a definite order. For example, on large motors or machines lubricated by a separate oil pumping system, it is essential that the lubricating system be in operation before the motor or machinery is started. For applications that require forced ventilation, the control system should be arranged so that the equipment can not be operated unless the ventilating system is functioning. This particular circuit shows a low-voltage protection scheme which is interlocked with another control or device. The interlocking contact must be closed before the start pushbutton is effective. If the interlocking contact should open while the control is in operation, the contactor coil *M* will be de-energized and the motor will be stopped. Such a scheme permits operation only when the first control or device is in operation and causes the interlocking contact to be closed.
5	D-C—Sequence Interlocking of Low-Voltage Release Motor Starters 	Low-voltage release is the term applied to a circuit where the control means, such as a maintained contact type of pushbutton, is used to energize the circuit at all times. When an attendant can not always be present at some applications, such as ventilating-fan or water-pump installations, this type of circuit is used so that the motor can be kept running at all times, or be ready to run whenever power is available. The same scheme of interlocking illustrated in *Fig.* 4 can be used to electrically interlock this circuit with another one which actuates the interlocking contact. With an LVR circuit, the overload-relay contact should always be of a type that requires manual resetting. For some overload conditions, an automatic resetting overload-relay contact will cause "pumping" and may eventually burn out the motor windings. Pumping is the term applied to the condition where a motor is repeatedly energized in a cycle controlled by the cooling and heating of the overload relay actuating means.

Fig.	Circuit Diagram	Description and Application
6	D-C—Motor Starting—Low-Voltage Protection Wide Operating Voltage Range 	Some applications require that a control circuit operate over an abnormally wide voltage range. Increased operational range can be secured by using a resistor in the auxiliary contact M_a holding circuit of an LVP control circuit. Practically all d-c contactors will remain closed on a relatively lower magnetizing force than that required initially to close the armature. Typical d-c contactors will close at approximately 60% of rated voltage when the coils are cold and will continue to hold the armatures closed when the energizing voltage is decreased to as low as 10% of rated voltage. By utilizing this characteristic, the contactor coil can be initially energized at normal or higher than normal voltage to close the contacts. When the start button is released, the protective resistor in series with the auxiliary contact M_a will become effective to limit the coil M energizing current to a value that will not cause overheating.
7	D-C Motor Starting—Field-Loss Protection 	A d-c shunt or compound-wound motor is usually damaged by overspeeding if the shunt-field excitation should fail during the starting or running period. This hazardous condition can be prevented by using a relay for indicating the loss of the field. The field-loss relay indicted by *FL* prevents the starting and operation of the control circuit unless there is current flow in the shunt-field circuit. The magnetic circuit of a typical field-loss relay is shown above the circuit diagram. One series-operating coil could be used, but, by using two coils, the current range of the relay can be approximately doubled. For this type of application, the field-loss relay must always be able to close at a current value which is less than minimum field current and be able to carry maximum field current without overheating the relay coils. This application characteristic is shown in the curve. With the field-circuit current flowing, the *FL* contacts in series with overload-relay contact *OL* and the stop button will close and the controller can be operated in the normal manner. Discontinuance of field current will de-energize the field-loss relay and the *FL* contacts will immediately open to stop the motor.
8	D-C Motor Starting—Field-Loss Protection 	Field-loss relay in *Fig.* 7 will prevent operation of the control circuit if there is no current flow in the field circuit. However, this type of current-operated relay will not provide protection in case part of the field is shorted out as the short-circuit current may still cause the field-loss relay to function normally. The addition of a differential field-loss relay *DFL* will provide protection against this type of field failure. One disadvantage to this particular protection scheme is that it requires an additional lead to be brought out at the midpoint of identical field sections. However, if the application is vital enough to require uninterrupted operation, this additional protective device is entirely justified. For differential field-loss protection the same type of relay as used for field-loss indication can be used, the only difference being in operating the relay with two identical potential coils instead of series coils. The potential coils *DFL* are connected across identical sections of the shunt field in such a way that the magnetic force produced by one coil is in opposition to that produced by the other coil. As the total net magnetic field is zero, the relay will not operate. However, if a short-circuit or winding failure occurs in one of the field sections, the voltage drop across that section of the field will change and the field produced by the other relay coil will be sufficient to operate the relay. This operation characteristic is illustrated in the sketch above the diagram. When the *DFL* relay operates, its normally-closed contacts opens to de-energize the line contactor coil *M* and stop the motor.

9 D-C Motor Starting—Manually-Operated Drum with Low-Voltage and Overload Protection

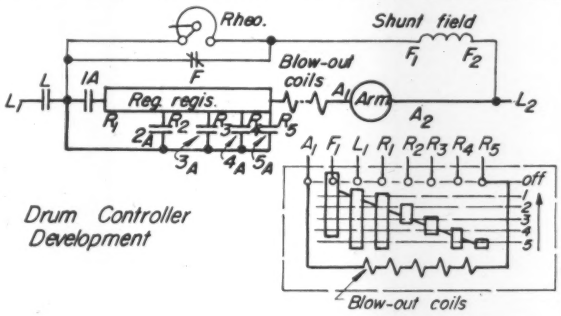


A manually-operated drum controller is constructed in such a manner that the sequence of starting and speed control is mechanically established by the arrangement of the sliding contacts or switches. This construction provides an exceptionally sturdy controller and prevents operation in other than the preset sequence. The circuit diagram shows a drum controller for starting and regulating the speed of a series motor by armature resistance control.

As a d-c motor should always be started with all the resistance in series with the motor armature, a line contactor *M* must be used to provide LVP operation. Also, as overload relays usually have contacts suitable for interrupting control circuit currents only, the line contactor must be used to obtain overload protection.

The contactor coil *M* is initially energized through a reset contact on the drum which is closed only in the off position of the drum. The auxiliary contact M_a provides a circuit for energizing the coil *M* when the drum controller is moved from the off position. Voltage failure or operation of the overload relay will de-energize coil *M* and the *M* contacts will then open the motor armature circuit to stop the motor. The contactor *M* can be reclosed only by returning the drum controller to its off position. This feature prevents starting of the motor unless total armature resistance is in the circuit.

10 D-C Motor Starting—Manually-Operated Drum with Provision for Full-Field Acceleration

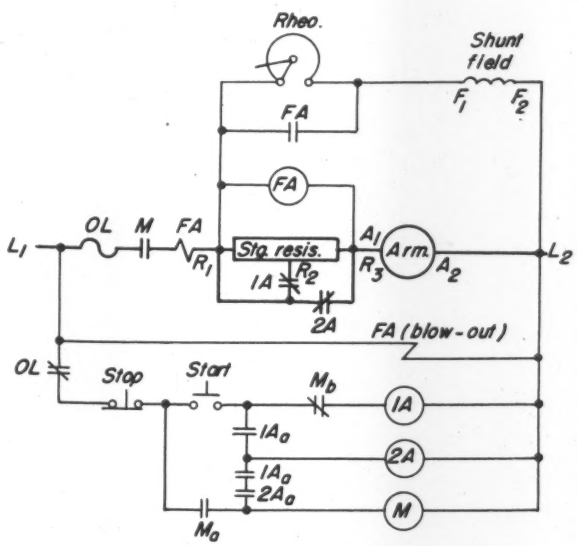


A d-c shunt or compound-wound motor should always be started with full shunt-field excitation. Also, the motor should generally be operated with full field when less than rated voltage is impressed upon the armature. As the torque developed by a d-c motor is directly proportional to product of the shunt-field flux and the armature current, the shunt field must be at its maximum strength for a minimum armature current.

This full field condition is readily accomplished on a drum controller by having a drum contact *F* which opens only on the last or 5th point of the drum. The field rheostat in series with the shunt field is effective only on the last drum point for reducing the shunt-field excitation and changing the motor speed.

For LVP operation and overload protection, a line contactor and a reset contact on the drum must be used as described for the circuit in *Fig.* 9.

11 D-C Two-Step Motor Starting With Provision for Full-Field Acceleration

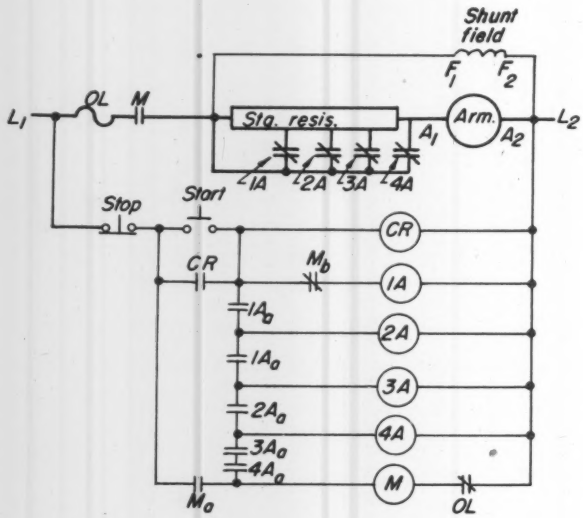
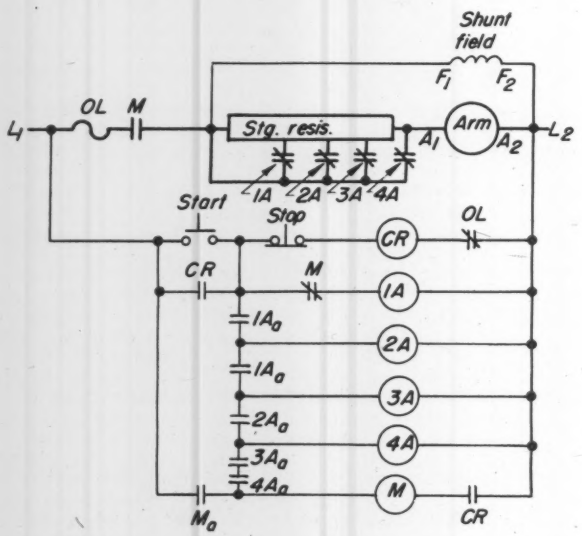


This is a diagram for a typical nonreversing adjustable-speed, shunt-wound motor control circuit. Timetactors 1*A* and 2*A* are combination timing relays and normally-closed accelerating contactors.

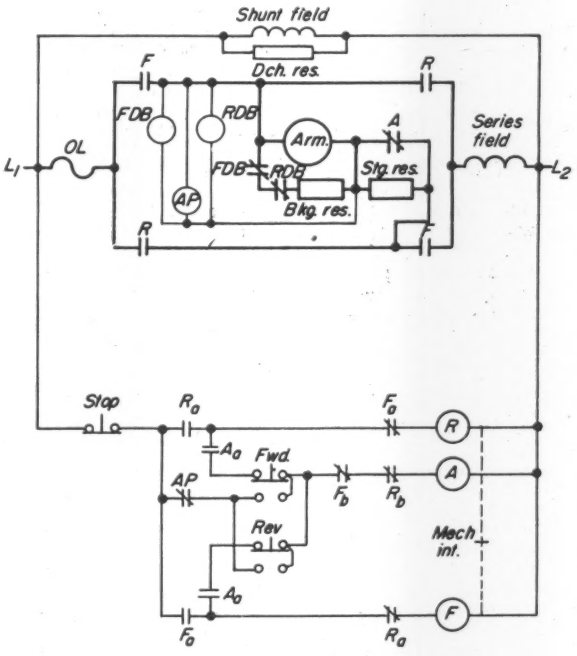
After the coils 1*A* and 2*A* of the Timetactors have been energized, the time-delay period can be initiated by de-energizing the coil circuits. The decay of the magnetic field produced by these coils induces a current in a copper tube or short-circuiting winding around the Timetactor core and provides a sustaining magnetic field for momentarily holding the Timetactor closed.

In starting a d-c motor it is imperative that all of the starting resistance be in series with the armature to limit the starting current to a safe value. Operation of the start button energizes Timetactor 1*A* which opens its main contact 1*A* across the first step of the starting resistor and closes its control contact $1A_a$ to energize the coil of the second Timetactor 2*A*. As the main contact 2*A* is now open, all the starting resistance is in series with the motor armature. Closing of the auxiliary contact $2A_a$ now completes the circuit to coil *M*. Contacts *M* close and energize the motor circuit. The line contactor coil *M* is energized and holds itself in through its auxiliary contact M_a. Also, the auxiliary contact M_b de-energizes the Timetactor coil 1*A* and initiates the timing period. After a suitable time delay, the Timetactor armature will be released and the main contact 1*A* will close and short out the first step of the starting resistor. Opening the control contacts $1A_a$ will now initiate the timing period of the second Timetactor. Now, all the starting resistance will be shorted out of the armature circuit and full voltage applied to the motor.

A field accelerating relay *FA* similar to the one used for field-loss protection is generally used to provide full shunt-field excitation during starting and any other period when the motor current exceeds a preset value. The potential coil *FA* in parallel with the starting resistor closes the relay contacts during the motor starting period as the drop across the resistor is approximately equal to line voltage less the motor armature *IR* drop plus counter emf.

Fig.	Circuit Diagram	Description and Application
12	D-C Four-Step Motor Starting Magnetically Operated 	Careful design of the circuit for a multistep, definite-time, d-c control is important to achieve the most reliable operation. For controls with a large number of contactors and relays, it is generally necessary to use auxiliary control relays as the currents involved are beyond the capacity of the pushbuttons and overload relay contacts. *Figs.* 12 and 13 compare two possible wiring schemes and illustrate how the proper electrical interlocking secures the most satisfactory operation. Timetactors 1*A*, 2*A*, 3*A*, and 4*A* operate similarly to those described for *Fig.* 11. Except for the addition of the *CR* relay and two more steps, this four-step control circuit operates exactly like the two-step starter previously described. This particular circuit will usually cause trouble during inching service as the line contacts *M* may weld together and cause difficulty in stopping the motor except by opening the power circuit by emergency means. If the stop button is operated before all the Timetactors have functioned to short out the starting resistance, the inductive discharge from these coils through the line contactor coil *M* may be sufficient to delay the opening of the contactor. Also, if the stop button is operated before the line contactor *M* contacts are completely sealed, the contact tips will weld together. This is caused by the delay in opening of the contactor from the Timetactor coil inductive discharge and by the inrush of the starting current through contacts which are held together with an abnormally low force. The time required to pick up all the Timetactors and to close the holding circuit contact M_a is relatively long for this multistep starter. With the *CR* relay it is necessary for the operator to hold the start button down only long enough to pick up the *CR* relay.
13	D-C Four-Step Motor Starting This Circuit Superior to one in *Fig.* 12 	By using a two-pole, normally-open *CR* relay instead of a single-pole relay and by wiring the control circuit as shown in *Fig.* 13, the previously described trouble during inching service can be prevented. With this circuit, the line contactor *M* will open immediately after the stop button is operated as the *CR* contact in the *M* coil circuit will prevent the other coils from discharging through it. Another advantage to this particular circuit is that the pushbuttons control only the *CR* relay. The pushbutton current carrying and interrupting capacity need be just sufficient to handle the *CR* relay. For the scheme illustrated in *Fig.* 12, the stop button must have an interruption rating sufficient to handle the total current required by all the coils in parallel. These two circuits illustrate particularly well how additional reliability and safety can be readily obtained by the addition of only one relay contact and a slight revision of the basic control circuit.

14 D-C One-Step Motor Starting with Provision for Reversing and Dynamic Braking



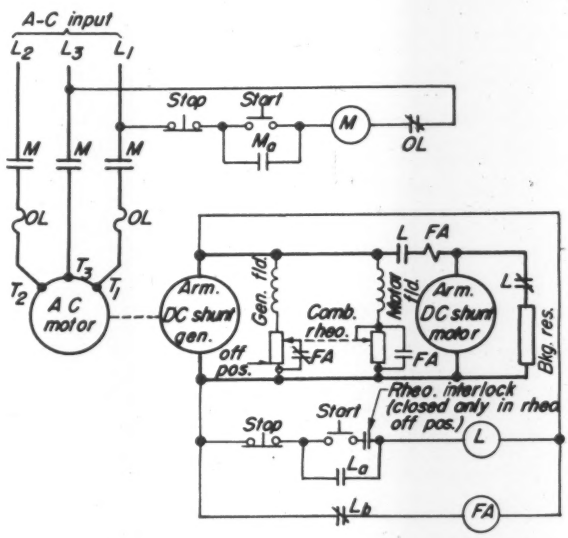
This circuit is similar to those used for one-step, d-c reversing starters for shunt-wound motors. The forward and reverse contactors for low-rating motors are frequently manufactured with two normally-open contacts on the upper end of the armature and one normally-closed contact on the lower end for dynamic braking. The contactor armature is pivoted at the middle and has an upper *R* or *F* and a lower *FDB* or *RDB* operating coil. This contactor construction mechanically interlocks the upper and lower contacts.

The forward and reverse contactors are customarily mechanically interlocked so that not more than one set of contacts can be closed at the same time. When the forward pushbutton is operated, Timetactor coil *A* will be energized and its main contact *A* will open to make the starting resistor effective. Its auxiliary contacts A_a will close and energize the forward *F* coil. The pushbuttons are interlocked in such a manner that pushing both buttons simultaneously will prevent either contactor *F* or *R* operating coil being energized. When the *F* coil is energized, the two normally-open forward directional contacts *F* will close and the forward normally-closed *FDB* dynamic braking contact will open. The forward contactor auxiliary contact F_b opens to de-energize the Timetactor coil *A* and initiate the time delay period. After a suitable time delay period, contact *A* closes and shorts out the starting resistor. The motor will then operate with full voltage on the armature.

As the lower contactor coils *FDB* and *RDB* are connected across the motor armature, they will be energized by the counter emf generated by the motor during the starting period. Even though these coils will be eventually energized with line voltage, the lower magnet pull will not be sufficient to operate the contactor as the upper coil pull is much greater with the armature in the sealed position. When the stop button is operated, the dynamic braking contact *FDB* will be snapped closed and held closed until the armature voltage is very low. Connecting the lower contact operating coils in this manner provides a fast closing of the braking contacts, good contact force and elimination of contact bounce, and prevents the opposite-direction contactor from being closed until the motor is almost at standstill.

With this type of forward and reverse contactor the antiplugging relay *AP* is not usually used. However, when the directional and braking contactors are separate devices, it is used to prevent the motor from being plugged. This relay should have a relatively high pick-up and a very low drop-out characteristic. The control contact *AP* in this circuit makes the forward and reverse buttons ineffective until it recloses when the motor armature is almost at standstill.

15 D-C Adjustable-Speed Drive with Armature Voltage and Field Control Manually-Operated Combination Generator and Motor Rheostat

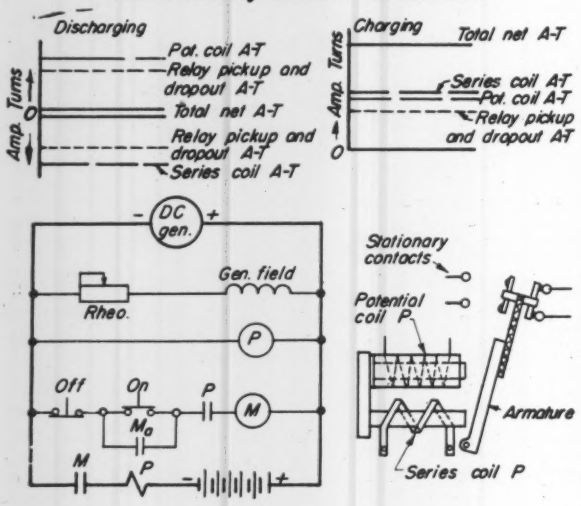
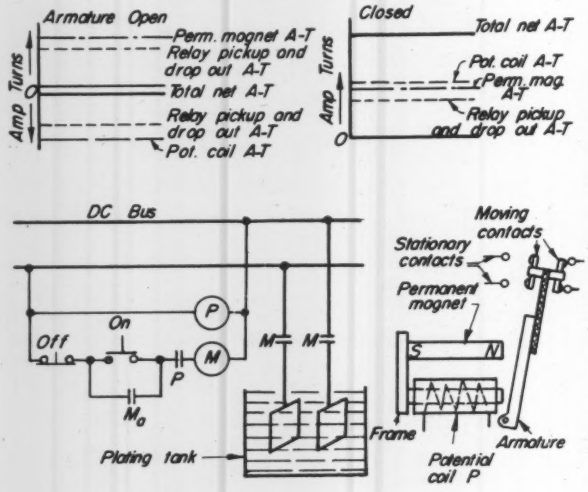
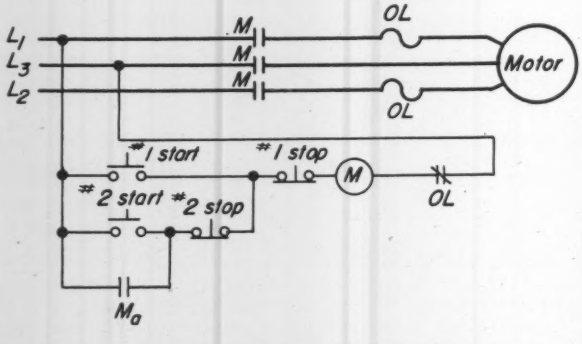


Adjustable-voltage drives provide speed ranges beyond that practical with a shunt-wound motor with field excitation only. This circuit is for a self-excited adjustable-speed drive.

When the start button in series with the linestarter coil *M* is operated, the a-c drive motor of the m-g set will be put in operation. The d-c output voltage of the generator can be controlled by means of the generator shunt-field rheostat. The speed of the d-c motor can be controlled by both variation of the voltage impressed across its armature and its shunt-field excitation. Operating the start button in the d-c circuit will operate contactor *L* which closes its normally-open *L* contacts and opens normally-closed, dynamic-braking contact *L*.

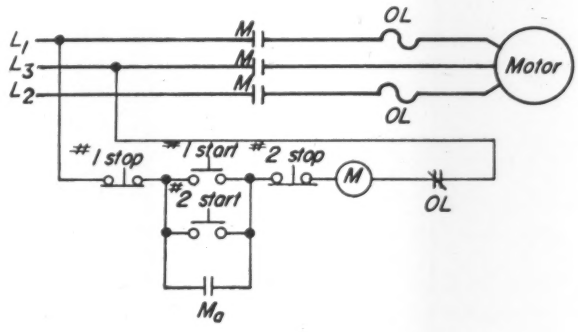
The generator and motor-field rheostat shafts are mechanically interlocked in such a manner that the voltage applied to the motor must increase to the rated value before the motor-field excitation can be decreased for speed increase by field control. The rheostat auxiliary contact, which is closed only in the rheostat complete-off position, prevents the motor from being started unless the rheostat is returned to its off position. This insures that the motor will start at its lowest speed.

The field accelerating relay *FA* operates in a manner previously described for *Fig.* 11 when the series coil *FA* is effective. If the d-c motor current increases above a preset value, the motor-field rheostat will be shorted out to provide full field and the generator-field rheostat will be cut into the circuit to decrease the generated voltage. The field accelerating relay contact will flutter until the conditions have stabilized. The shunt coil *FA* also operates the *FA* contacts when the d-c motor is not running. This arrangement reduces the generator excitation to a minimum and provides full-field excitation on the motor for effective dynamic braking.

Fig.	Circuit Diagram	Description and Application
16	**D-C Battery Charging with Polarity-Control Protection** 	For polarity control of a battery-charger control circuit, the same type of relay used for field-loss protection and field-acceleration control can be used. This particular application requires a potential coil and a series coil which are connected so that their magnetic effects are cumulative for normal operation. When the generator is in operation, potential coil *P* will be energized and the control contact *P* will be held closed. The contactor *M* operates when the on button is depressed. If the charging current direction is correct, the magnetic field developed by the series coil *P* will help to hold the relay armature closed. This relay operation characteristic is illustrated by the sketch above the circuit diagram. If the generator voltage should become less than that of the battery, the charging current would reverse and tend to drive the generator as a motor. This condition would rapidly discharge the battery. Reversal of the current through the series coil neutralizes the magnetic effect of the potential coil and allows the relay armature to open and de-energize the battery circuit. This relay magnet characteristic is shown above the circuit diagram. If the generator driving motor should stop, this polarizing control immediately functions to open the *M* contacts and prevent the battery from discharging.
17	**D-C Electro-Chemical Processing with Polarity-Control Protection** 	In many electro-chemical processes a reversal of current may cause damage or trouble even if the current has a small magnitude or is of short duration. A relay with a permanent magnet in the magnetic circuit can be used as a polarizing relay. A potential coil *P* indicates the polarity of the d-c bus. If the polarity of the bus is correct, the relay armature will close the contacts *P* and makes the pushbutton effective to close the *M* contacts. The direction of the current flow in the coil *P* either assists or neutralizes the permanent-magnet field. The relay armature is either attracted or repelled depending upon the coil polarity. These two operating conditions are graphically illustrated by the sketches above the circuit diagram. This particular relay could also be used to control the polarity of a d-c generator and prevent operation unless the polarity is correct. It could also be used to prevent a d-c machine tool motor from operating except in the desired direction.
18	**A-C Full-Voltage Motor Starting with Two 3-Wire Pushbutton Stations** 	Many machines require more than one control station for most convenient operation. Low cost and standard duty start-stop pushbutton stations are frequently constructed mechanically in such a manner that the normally-open and normally-closed units have a common point and it is almost physically impossible to separate the electrical connection between the two units. Adapting this station or more than one of these stations to a control circuit is sometimes difficult because of the common-point mechanical construction. This circuit diagram shows two 3-wire start-stop stations applied to an a-c linestarter. This particular scheme requires only three wires to be run to the stations from the control panel. However, this circuit has a serious defect as the No. 1 start button is still effective even though the No. 2 stop button may be held down. Such a wiring scheme might cause an accident as the operator might depress the No. 2 stop button and believe the machine safe to repair or adjust. In this case, if the No. 1 start button is operated, the motor will start and may injure the operator.

A-C Full-Voltage Motor Starting with Two 3-Wire Pushbutton Stations
This Circuit Superior to one Shown in *Fig.* 18

19

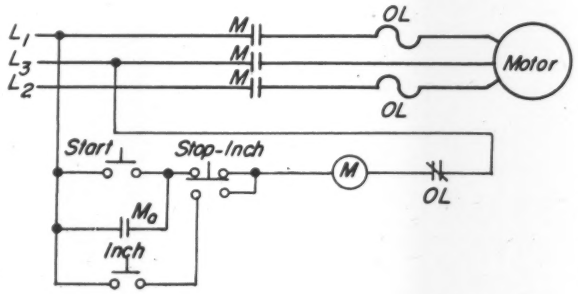


The identical apparatus of *Fig.* 18 may be wired in the circuit as shown by this diagram and the objectionable feature can be eliminated. This circuit requires four wire connections between the control panel and the pushbutton stations. The expense of running an extra wire is a small price to pay for complete safety.

The stop button should always be supreme for controlling the starter and should not be by-passed at any time if complete safety is to be secured. More than two of these 3-wire stations can not be safely applied to a circuit.

A-C Full-Voltage Motor Starting with Provision for Inching

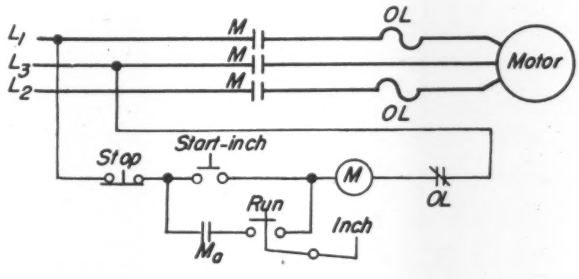
20



Many machines require "inching" or operation of the drive motor for periods only as long as the inch pushbutton is held down. This circuit diagram shows an inching circuit in which the inch button is effective only when the stop-inch button is held down. Holding the stop-inch button down disrupts the *M* coil holding circuit and provides a circuit from the inch button to the coil. A disadvantage to this circuit is that two fingers must be used to operate the two buttons to accomplish the inching operation. If the stop-inch button should be released slightly before the inch button, the upper contacts of the stop-inch button may be closed before the auxiliary contact M_a has opened. This will re-energize the contactor coil *M* and permit the motor to operate even after the inch button has been released. If the contactor parts are massive and relatively slow in operation because of the inertia, this false operation is almost certain to occur. It is particularly likely if the stop-inch button moving parts are light and fast in operation.

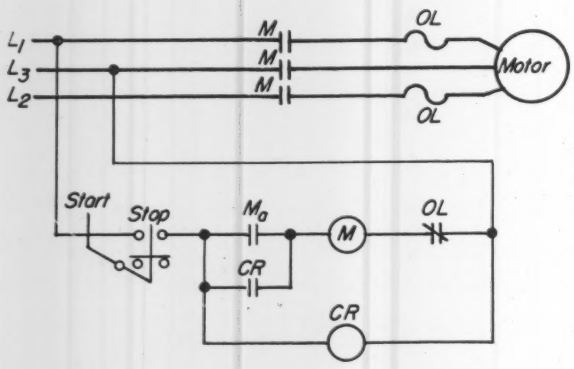
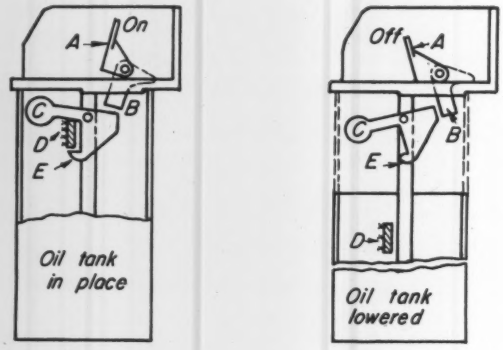
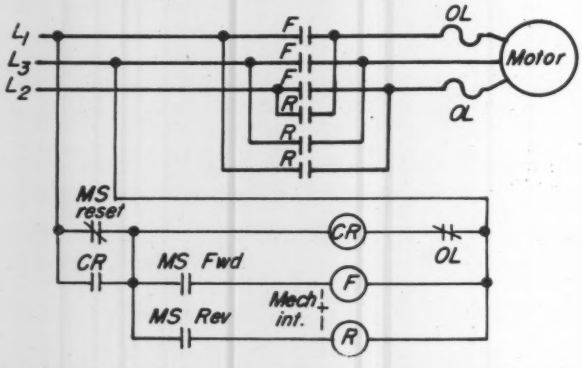
A-C Full-Voltage Motor Starting with Provision for Inching
This circuit superior to one shown in *Fig.* 20

21



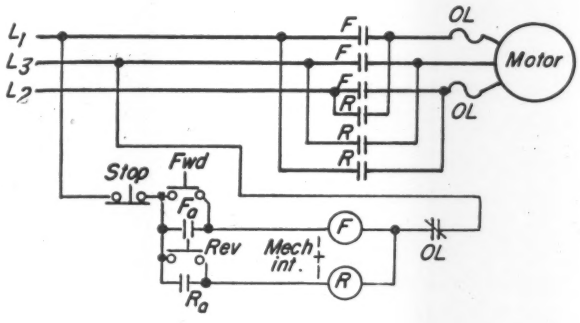
The undesirable characteristics of the wiring scheme of *Fig.* 20 may be corrected by using a maintained type of run-inch pushbutton in the *M* coil holding circuit. After the run-inch button has been set for the operation desired, the start button may be operated for starting or inching by only one finger. This satisfactory control scheme can be obtained by adding only a latching pushbutton unit to the apparatus required for the circuit in *Fig.* 20.

For fully automatic inching operation, a control relay must usually be added to the circuit. With a separate relay for inching, it is necessary to operate only the inch button for inching. This particular scheme is not shown.

Fig.	Circuit Diagram	Description and Application
22	A-C Full-Voltage Motor Starting with Protection Against "Pumping" for a Low-Voltage Release Control Circuit 	When an application requires a low-voltage release control circuit and when it is impossible to obtain a manually reset overload relay, a special control relay *CR* can be used to prevent "pumping" of a motor because of an overload. Also, the overload relay, or the control panel may be located where it is inconvenient to manually reset a relay. When the *CR* relay coil of this circuit is energized by operation of the start button, its armature moves to the sealed position and the *CR* contacts close for a short period of time and then open. The *CR* contacts are closed a sufficient length of time for the *M* coil to be energized and for the auxiliary contacts M_a to close and provide a holding circuit. In case of a power failure, the *CR* relay repeats the starting cycle automatically. However, in case of an overload the *M* contactor coil is de-energized and the *M* contacts open to de-energize the motor. If the overload relay is of the automatic-reset type, the *M* contactor coil will not be energized even when the *OL* contacts reclose because the *CR* and M_a contacts are open. The stop button must be operated to de-energize the *CR* relay coil and prepare it for another starting cycle. In oil immersed equipment, the *CR* relay is sometimes equipped with a dash-pot mechanism to provide a momentary time delay for the period that the *CR* contacts are closed. The armature of the *CR* relay remains in its sealed position as long as the coil *CR* is energized.
23	A-C Full-Voltage Motor Starting for Hazardous or Corrosive Atmosphere Conditions 	Oil immersed linestarters are frequently applied for across-the-line starting of squirrel-cage motors where the atmosphere may contain explosive or corrosive gases. For corrosive atmosphere applications the arcing parts are immersed under at least a two-inch head of oil. For Class I, Group D hazardous locations (explosive atmosphere), the electrical equipment is immersed under at least a six-inch head of oil. This type of starter is often equipped with a circuit breaker or a disconnecting switch which is also immersed in the same oil tank. The disconnecting switch is operated by means of a mechanical linkage to an operating handle on the outside of the starter. This sketch shows the mechanical interlocking that is provided between the tank and the disconnecting switch to prevent any arcing when the electrical apparatus is not immersed in oil. When the breaker handle *A* is in the on position the lever *B* prevents the latch *E* from swinging out of the way and permitting the lug *D* on the tank from being moved downward. This prevents the tank from being removed when the disconnecting switch is closed. When the tank has been removed, the weight *C* places the latch arm in such a position that the handle *A* can not be moved to the on position.
24	A-C Full-Voltage Motor Starting with Provision for Reversing Manually-Operated Master Switch 	When two-wire master switches or pilot devices are used, it is necessary to use a control relay *CR* to provide low-voltage protection control and overload protection. In this circuit a manually operated master switch *MS* is used to select the motor operation in either the forward or reverse direction. The master switch reset contact *MS*, which is closed only in the master switch off position, provides a circuit for initially energizing the *CR* relay coil. Relay *CR* holds itself in the circuit through its contact *CR* when the master switch is moved to its forward or reverse position. When the master switch is in the forward or reverse position, the reset switch will be open and all power for the control circuit will be supplied through the *CR* contacts. In case of an overload or a voltage failure, it is necessary for the operator to return the master switch handle to the off position to re-energize the *CR* relay coil and the control circuit before the motor may be restarted.

25

A-C Full-Voltage Motor Starting with Provision for Reversing

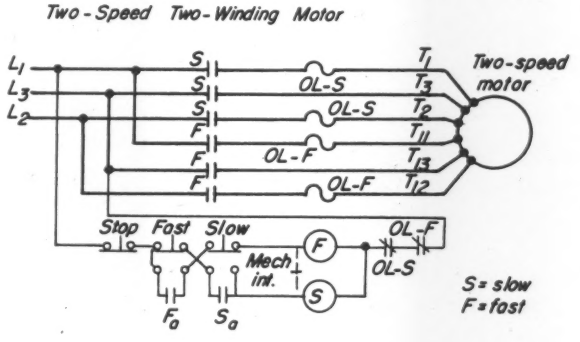


Mechanical interlocking system are usually provided on reversing control circuits to prevent short-circuits between phases of the power supply. A mechanical interlocking device between the forward and reverse contactors prevents more than one contactor from being closed at the same time even though both contactor coils may be energized.

This circuit illustrates a possible wiring scheme for low-voltage protection. With this pushbutton circuit it is possible to energize both the forward *F* and reverse *R* contactor coils at the same time if the two buttons are operated simultaneously. Under such operation, the rotation of the motor depends only upon which contactor closes first and is unpredictable. Such operation also places an unnecessary strain on the mechanical and electrical apparatus. Also, when the motor of this circuit is being operated, it is necessary to operate the stop button before the opposite-direction pushbutton and contactor become effective. Safety with this circuit depends wholly upon the mechanical interlock.

26

A-C Full-Voltage Motor Starting with Provision for Reversing

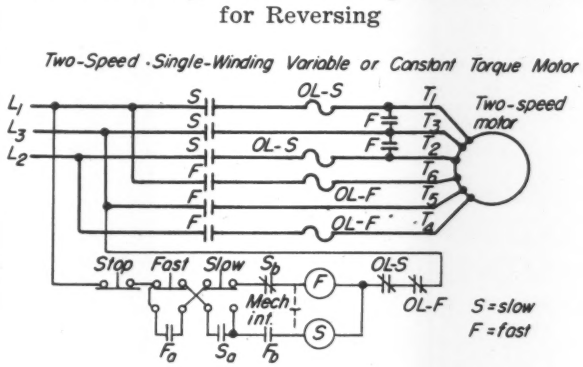


The electrical apparatus required for controlling a two-speed, two-winding multispeed motor is practically the same as that for an ordinary reversing circuit, *Fig.* 25. These control circuits can be and often are identical for reversing or multispeed duty.

This circuit is similar to the one shown in *Fig.* 25 except the pushbuttons are wired differently. No additional apparatus is required. This circuit is superior to the previous one because it is impossible to energize either the *F* or *S* coil if both the fast and slow pushbuttons are operated at the same time. Also, when changing direction of the rotation, it is not necessary to operate the stop button; the holding circuit to the energized coil is automatically interrupted when the desired direction button is operated. This particular wiring scheme requires two more wires from the pushbutton station to the control panel over that required for *Fig.* 25.

27

A-C Full-Voltage Motor Starting with Provision for Reversing



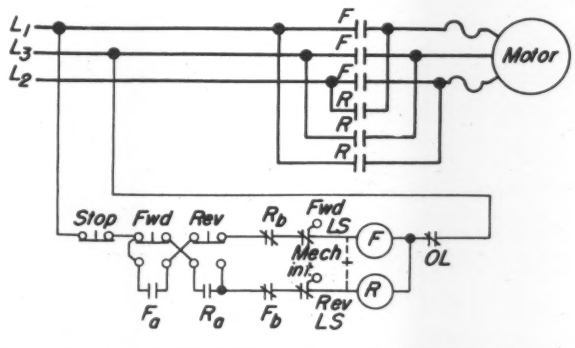
A two-speed, single-winding multispeed motor requires a three-pole and a five-pole mechanically-interlocked contactor. However, the control circuit is the same as used for almost any reversing or two-speed motor controller.

The addition of a normally-closed auxiliary contact to each of the line contactors provides additional electrical interlocking. One contactor coil can not be energized unless the other contactor is completely open. The opposite-direction contactor-coil circuit is always open as long as the other contactor is closed or even partially closed.

This particular scheme is the most desirable of the three schemes shown in *Figs.* 25, 26, and 27 because it not only provides additional safety of operation, but also reduces wear and strain on the mechanical interlock.

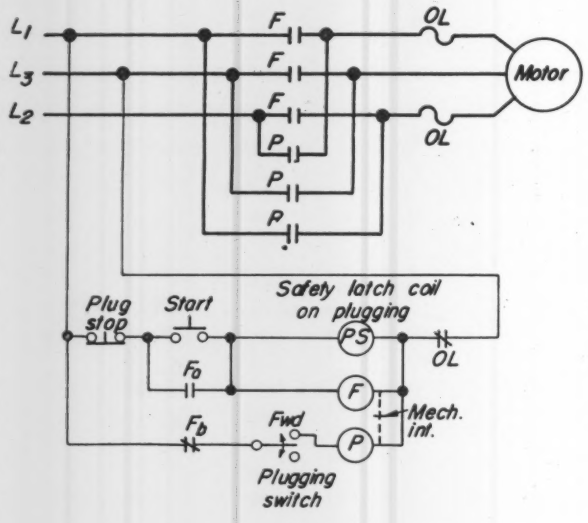
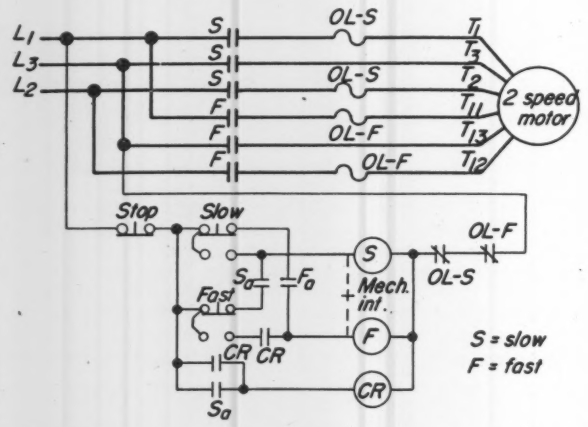
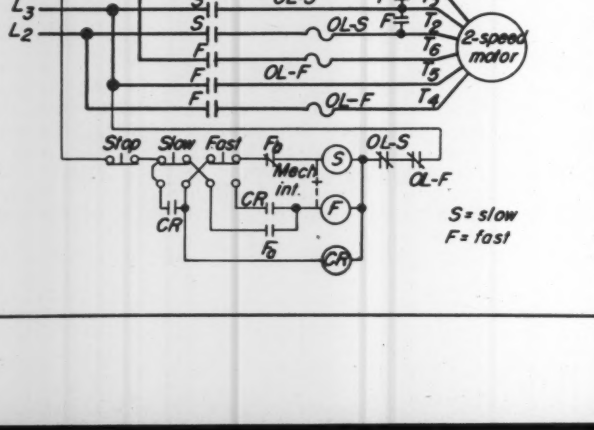
28

A-C Full-Voltage Motor Starting with Provision for Reversing and Limit-Switch Protection



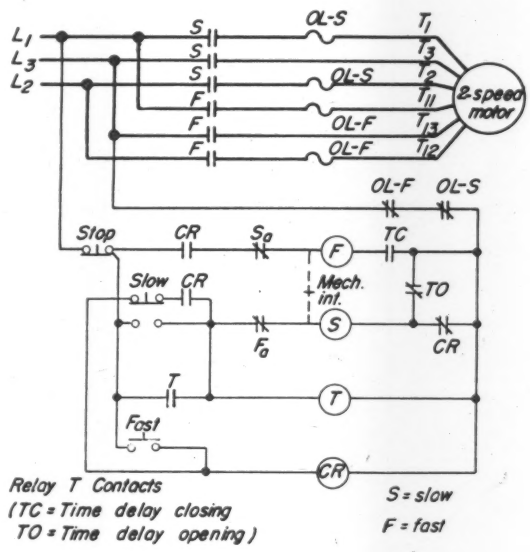
Many applications, such as hoists and machine tools, require that the motor must be stopped after the work, the machine or its parts have reached a certain position. Limit switches provide this co-ordination between the mechanical position and the functioning of the electrical equipment. Limit switches are usually simple contact devices which are actuated through levers or cams on the machine. After the limit switch for one direction has operated, it is impossible without special by-pass switches to operate further in that direction as the line contactor coil can not be energized. The limit switch may be reclosed by operating the motor in the opposite direction.

In this circuit the limit switches function to interlock the mechanical and electrical operation of the machine.

Fig.	Circuit Diagram	Description and Application
29	**A-C Full-Voltage Motor Starting with Provision for Plugging to a Stop** 	In some applications, production output may be increased by decreasing the time required for the spindle of the machine to come to rest after the stop button has been operated. A zero-speed switch can be utilized for stopping the motor in a minimum of time. The plugging switch illustrated in this circuit is centrifugally operated. The switch is direct-connected to the motor or at a convenient place on the machine. A fly-ball mechanism operates to close the contacts when the motor attains a certain speed. The contacts will remain closed until the speed drops to a preset value. Typical plugging switches can be adjusted so that the contacts will open in a range from 500 to 100 rpm. With this circuit the motor can be started in the normal manner for a LVP circuit and the motor will operate in its correct direction. When the plug-stop button is operated, the contactor coil *F* will be de-energized and the normally-closed auxiliary contact F_b in series with the plugging switch and plugging contactor coil *P* will close. The contacts *P* now close to energize the motor for the reverse direction of rotation. When the motor decelerates to the desired speed, the plugging switch opens to discontinue plugging the motor. When properly set, the motor will not coast past the standstill position. The safety latch coil *PS* on the plugging switch prevents accidental starting. When the LVP circuit is de-energized, the safety latch magnet prevents the plugging switch contacts from closing. Only when the latch coil is energized, or if the motor is already operating, can the switch contacts close or continue to stay closed.
30	**A-C Full-Voltage Two-Speed Motor Starting with Provision for Compelling Operator to Start on Slow Speed** *Two-speed Two-winding Motor* 	Some applications of multispeed motors require that the motor always be started on the lowest speed in order not to damage the motor or driven load. A relay applied to the control circuit to enforce such an operating sequence is known as a "compelling relay." As shown by this circuit diagram, the fast contactor coil *F* can not possibly be energized until the *CR* contact on the compelling relay *CR* is closed. The *CR* relay is energized initially when the slow contactor auxiliary contact S_a closes. Once the *CR* relay coil is energized, it holds itself in the circuit through one of its own normally-open *CR* contacts. The *CR* relay remains energized until the stop button is depressed. Once the *CR* relay is energized, the motor can be operated on the fast speed by pushing the fast button which de-energizes the slow contactor coil *S* and energizes the fast-speed contactor coil *F*. This particular circuit has one defect that may cause false operation. If the motor is operating at either the fast or slow speed, bumping or partially depressing a fast or slow button will open the coil holding circuit and stop the motor. As the compelling relay is not dropped out, the motor can then be started on either the slow or fast speed. For this circuit to operate satisfactorily, the pushbuttons must always be fully depressed when changing from one speed to another.
31	**A-C Full-Voltage Two-Speed Motor Starting with Provision for Starting on Slow Speed** *Two-speed Single-Winding Variable or Constant Torque Motor* 	Elimination of one auxiliary contact S_a and a revision of the wiring shown in *Fig.* 30 will provide a much more satisfactory circuit. In this circuit the compelling relay coil *CR* is energized when the slow button is operated. The *CR* contact in parallel with the slow button provides a holding circuit for the *CR* coil and the slow-speed contactor coil *S*. Once the motor has been started on the slow speed, the fast button is effective for energizing the fast-speed contactor coil *F*. When the fast button is operated, coil *S* is de-energized and coil *F* is energized. While it is possible to change from fast to slow speed by partially depressing or bumping the slow button, the motor will not stop. When the stop button is operated to stop the motor, the *CR* relay is de-energized and is reset for compelling the next starting operation on slow speed.

A-C Full-Voltage Two-Speed Motor Starting with Provision for Automatic Timed Acceleration

32



While a compelling relay requires the operator to always start on the slow speed, it does not prevent him from immediately going to the fast speed. This circuit is for automatic acceleration with operation on the slow speed for a preset time period.

A motor-operated or a dash-pot, time-delay relay can provide the delay between slow and fast speeds. In this circuit the timing-relay operating coil is designated by *T*. The *T* contact closes immediately when the coil *T* is energized. The *TO* contact opens after a time delay period and the *TC* contact closes.

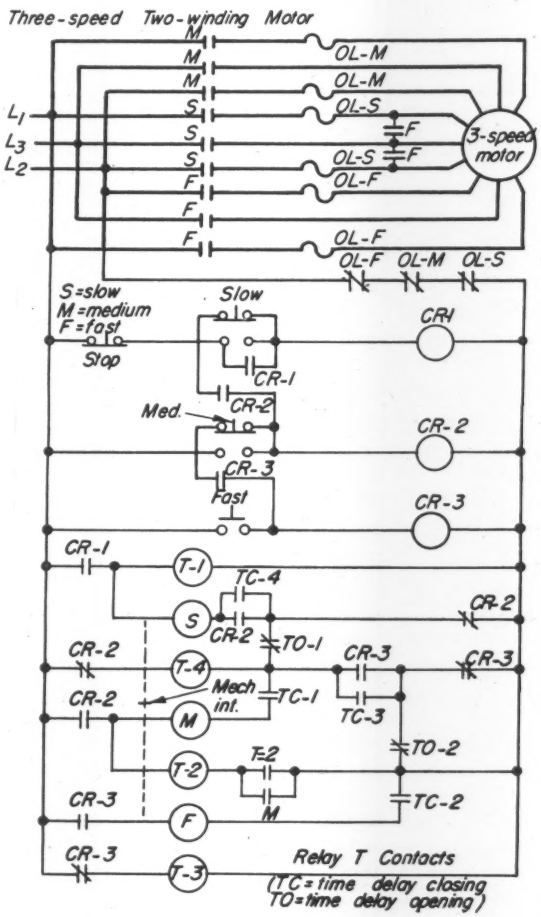
If the fast button is operated first, the *CR* relay will be picked up and will hold itself in the circuit through the upper contacts of the slow button, its own normally-open *CR* contacts, and the instantaneous timing-relay contact *T*. The slow-speed contactor will be closed and the timing relay *T* will be energized to start the timing period. After a definite time delay, contacts *TO* will open to de-energize the slow-contactor coil *S* and the *TC* contacts will close to energize the fast-contactor coil *F*. The fast-contactor coil can not be energized until the slow-contactor auxiliary contact S_a closes. After this, the operator may run on slow or fast speed without delay.

If the operator depresses the slow button first, the motor will start on the slow speed and timing relay period will be initiated. If the fast button is operated before the time period has elapsed, the motor will continue to run at the slow speed until the preset time has elapsed; then, the control will automatically change over for fast-speed operation.

For a preset deceleration period when changing from fast to slow speed, another timing relay is required. For a time delay period for changing from fast to slow, and then slow to fast, the circuit must be arranged so that the timing devices are de-energized and then re-energized.

A-C Full-Voltage Three-Speed Motor Starting with Provision for Automatic Timed Acceleration and Deceleration

33



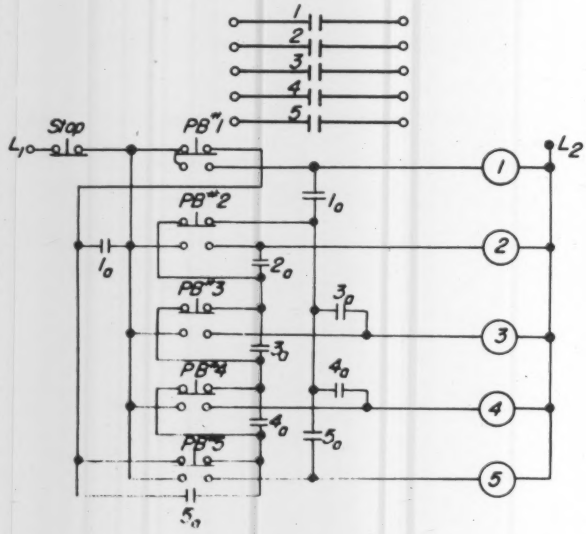
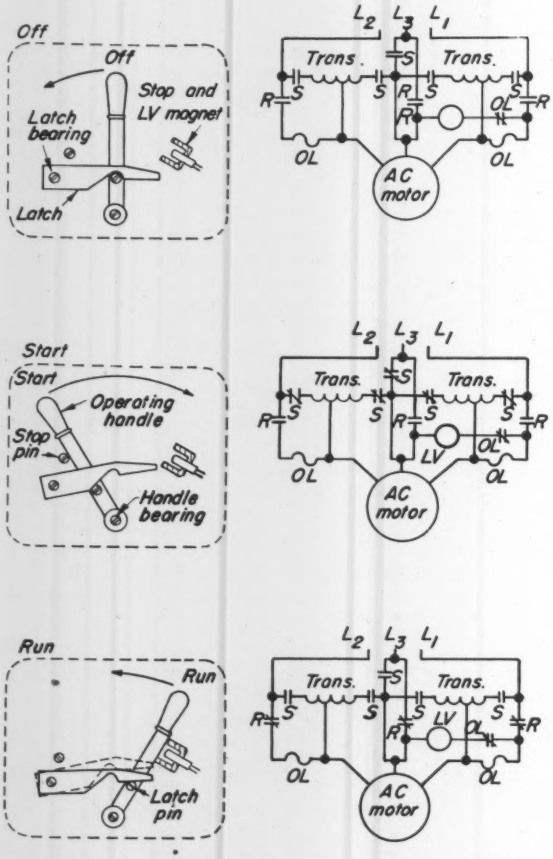
Some applications require timed automatic deceleration as well as timed automatic acceleration. A separate timing relay is necessary for each acceleration, deceleration, and speed change. This particular circuit is for a three-speed fan motor which requires timed acceleration and deceleration to prevent breakage of the shaft or fan blades.

When the control circuit is initially energized, timing relay coils *T*-3 and *T*-4 are energized and their timing periods start. Eventually contacts *TC*-3 and *TC*-4 close. The motor can not be started on the slow speed until the *TC*-4 contact has closed.

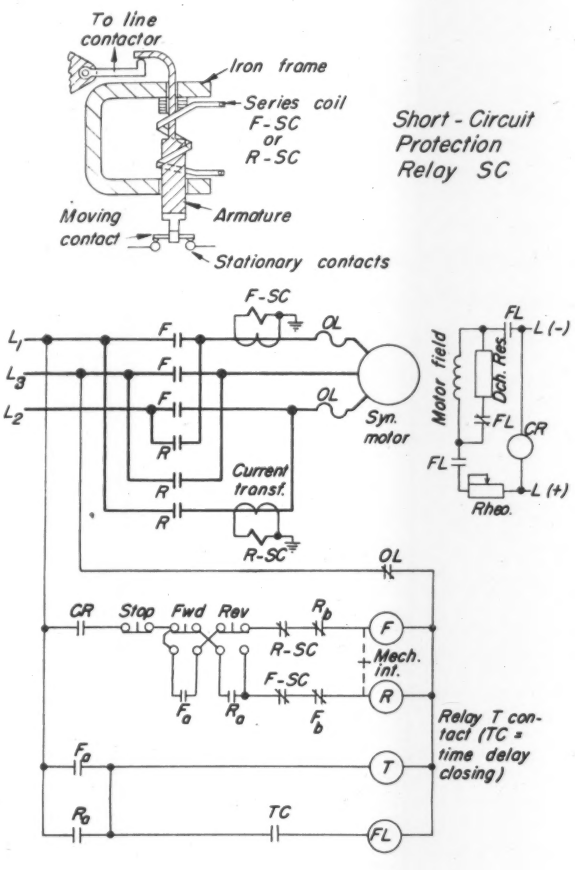
Operation of the fast button will automatically bring the motor up through the slow and medium speeds to the fast speed with a definite period of operation on each intermediate speed. The *T*-1 and *T*-2 timing relays are for acceleration and the *T*-3 and *T*-4 timing relays are for deceleration.

In decelerating from fast to slow speed, the medium-speed winding is not energized, but the time delay period of both relays *T*-3 and *T*-4 must elapse before the slow-speed winding can be energized.

The circuit that the pushbuttons control is interesting in that the number of control relays operated is accumulative. The slow button operates the *CR*-1 relay; the medium button operates the *CR*-2 and *CR*-1 relays; and the fast button operates the *CR*-3, *CR*-2, and *CR*-1 relays in that sequence.

Fig.	Circuit Diagram	Description and Application
34	**A-C or D-C—Selecting and Operating only one of a Number of Relays or Contactors** 	Multispeed control and special circuits sometimes require that only one of a number of relays be in operation at any one time. This particular circuit uses momentary contact pushbuttons for part of the interlocking. The basic principle of the scheme is that the particular relay selected must have its coil holding circuit through the upper contacts of all other pushbuttons except its own, and a by-pass contact for providing a circuit while its own pushbutton contact is returning from the lower to the upper contacts after being released. If one relay is energized, operation of any other pushbutton will de-energize it and select the particular relay desired. Mechanical interlocking and additional electrical interlocking may be added for a particular application. It should be noted that with the exception of the wiring for the first, second, and last relay the scheme is symmetrical and any number of relays may be added and wired the same as the third and fourth relays.
35	**A-C Reduced-Voltage Motor Starting Manually Operated with Provision for Overload and Low-Voltage Protection** 	The sketches and diagrams illustrate how a manually-operated, reduced-voltage starter for a-c motors can be completely mechanically interlocked to provide the proper starting sequence. Autostarters which use autotransformers for reducing the voltage applied to a motor are for application wherever the starting current of the motor is likely to exceed local power company restrictions, or interfere with plant operation. This type of control keeps the current inrush within specified limits and provides the maximum starting torque for least inrush current. For this particular type of controller, the start *S* and run *R* contacts are operated from a common shaft and are arranged similar to a double-throw switch. The starting operation requires five contacts and the running condition requires three contacts. With this arrangement it is impossible for both the starting and running contacts to be closed at the same time. START: Operating handle is moved in a counter-clockwise direction to close the *S* contacts and connect the autotransformer to the line and the motor to the reduced-voltage taps. Latch mechanism prevents the handle from moving in any direction except the start direction. To prevent operation too long on the starting windings, the handle will snap back to the off position if the handle is released. A spring return and centering mechanism tends to return the handle to the off position at all times. RUN: Pushing the operating handle in a clockwise direction will transfer the motor connections directly to the line through the *R* contacts. The operating handle must be moved rapidly or the latch will drop down by its own weight and prevent the handle from being moved to the run position. In the run position overload protection is provided and a magnet energized by the *LV* coil holds the operating handle in the run position. This magnet provides LVP and overload protection. STOP: Handle is held in the run position by the *LV* magnet until the operator pulls it to the off position, or until an overload or voltage failure de-energizes the *LV* magnet. The spring-return mechanism snaps the handle to the off position.

36 A-C Full-Voltage High-Voltage Motor Starting with Provision for Reversing and Short-Circuit Protection



Special current-operated relay can be used to prevent closing of an opposite-direction contactor until the arc is fully extinguished. On high-voltage a-c circuits, arcs are sometimes relatively difficult to extinguish. In some cases the contactor may be fully open and yet the arc current still energizes the motor windings. Ordinary electrical and mechanical interlocks can not prevent short-circuits to the power supply under such conditions.

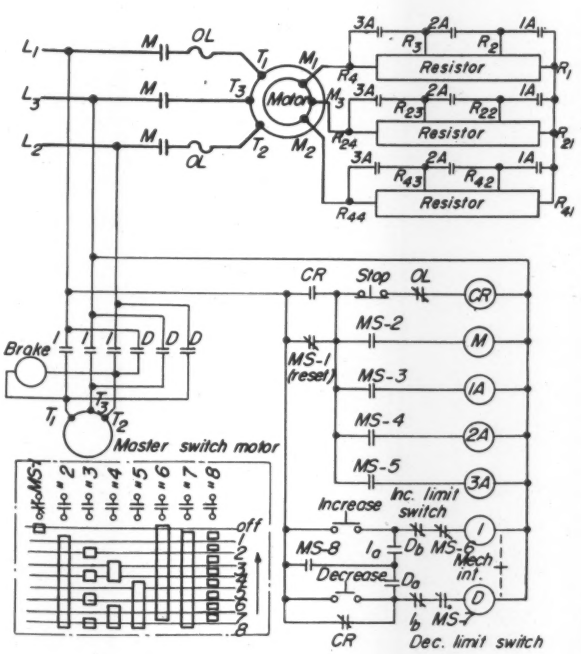
The sketch above the diagram illustrates a typical relay used for this purpose. The series coil *F-SC* or *R-SC* can not develop enough magnetic force to lift the armature and open the control contacts. One relay must be used for each directional contactor and a linkage on the contactor lifts its own relay armature to the closed position whenever it operates. Once the series coil is energized, the relay armature will remain in the closed position until the current through the coil drops to practically zero. With this protective means the arc current must be extinguished before the control contact will reclose to permit energizing the opposite-direction contactor. When a contactor opens, the relay armature is released mechanically; but it is held magnetically as long as there is current flow.

This circuit diagram shows two short-circuit protective relays applied to a high-voltage synchronous motor. The *CR* relay in the *d-c* power supply for the motor field prevents starting unless d-c power is available. It will also stop the motor in case the d-c power fails. It provides the interlocking between the a-c and d-c circuits.

When the forward button is operated, the *F* coil is energized. The *F* contacts close and start the motor in a forward direction. The *F-SC* contacts in series with coil *R* for the reverse direction are opened mechanically by the operation of the *F* contactor. These *F-SC* contacts can not be reclosed until the motor current and arcing current decrease to zero.

For simplicity of the circuit, the d-c field is applied after a definite period of time. Some of the modern synchronous motor control circuits function with respect to the current and speed of the motor for applying the field. The slip frequency in the d-c circuit is a perfect indication of the motor speed.

37 A-C Wound-Rotor Motor Starting and Speed Control

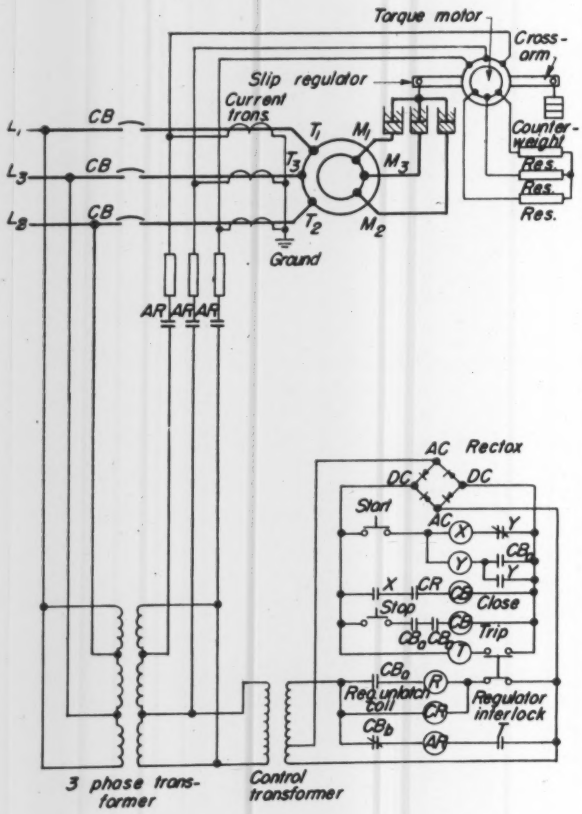


This circuit is for a wound-rotor motor control circuit which provides eight different operating speeds for only three contactors and three steps of resistance in the motor secondary circuit. The heart of the control system is the master switch which selects the energization of the 1*A*, 2*A*, and 3*A* contactors in a certain sequence. The secondary resistor is proportioned in a geometric ratio. The resistance of the first step has a ratio of 1, the second 2, and the third 4. In the first master switch position, seven units of resistance are in the circuit. In the second position, contactor 1*A* closes and leaves six units of resistance in the circuit. In the third position, contactor 2*A* only closes and cuts out two units of the resistance.

A "notching" contact *MS*-8 which closes on all master switch positions except at the precise point prevents this particular motor-operated master switch from being moved to and stopped at a position between the exact drum points. Once the increase or decrease pushbutton has been operated, the *MS*-8 contact and the I_a or D_a auxiliary contact will drive the master switch to the next definite point even though the pushbutton may be released.

When the stop button is operated, when the voltage fails or when the overload relay functions, the normally-closed *CR* contact will energize the *D* coil and return the drum automatically to its off position as soon as the power is available. The increase and decrease limit switches prevent driving the drum past the extreme drum positions.

This control is usually feasible only when a large number of speed points are required. With four secondary contactors, 16 speed points can be obtained; with five secondary contactors, 32 points can be obtained, etc.

Fig.	Circuit Diagram	Description and Application
38	A-C Wound-Rotor Motor Starting and Speed Control for Fly-Wheel Load Applications 	Large a-c oil circuit breakers are generally operated by d-c closing and tripping solenoids. As the solenoid coils are intermittently rated, special control schemes must be used to prevent burning out the coils. This circuit shows a primary circuit breaker *CB* applied to a wound-rotor motor which has a slip regulator for flywheel drive applications. The regulator interlock is actuated by the slip regulator crossarm and prevents the primary circuit breaker from closing unless there is the maximum resistance in the motor secondary circuit. When the start button is operated, relay *X* operates and energizes the *CB* closing solenoid. The breaker now closes and the CB_a contacts close to energize the regulator arm, unlatch coil *R* and energize coil *Y*. Relay *Y* operates to de-energize coil *X* regardless of how long the start button is held down. When the regulator crossarm is unlatched, the moving electrodes descend in the electrolyte and accelerate the motor. The interlocking scheme with relays *X* and *Y* prevent the closing solenoid *CB* from being energized any longer than is necessary to properly close the breaker. The main breaker contacts are then held closed by a mechanical latching device. When the stop button is operated, the *CB* trip coil is energized to open the *CB* contacts. Here again, the auxiliary breaker contacts CB_a open to de-energize the trip coil. This particular circuit is arranged so that when the main breaker opens, coil *AR* is energized for automatically returning the regulator crossarm to its off position of maximum resistance. This is accomplished by energizing the regulator torque motor from the three-phase transformer. The regulator interlock acts as a limit switch and time-delay relay *T* will de-energize the *AR* contactor after a short time delay from the instant that this interlock operates. This type control circuit is used with motors of 1000 to 7000 hp where it is desirable to permit a flywheel to give up its energy during sudden peak loads. The circuit breaker is generally equipped with an undervoltage release mechanism so that the breaker is tripped when the power fails. This particular feature is not shown.

Extensive electrical and mechanical interlocking is required in the Newton planer type milling machine shown in *Fig.* 40 to provide reliable operation and to prevent damage to the machine. This particular machine has fourteen a-c and d-c motors to drive the various components, each of which is protected against overtravel and interference with the motion of another.

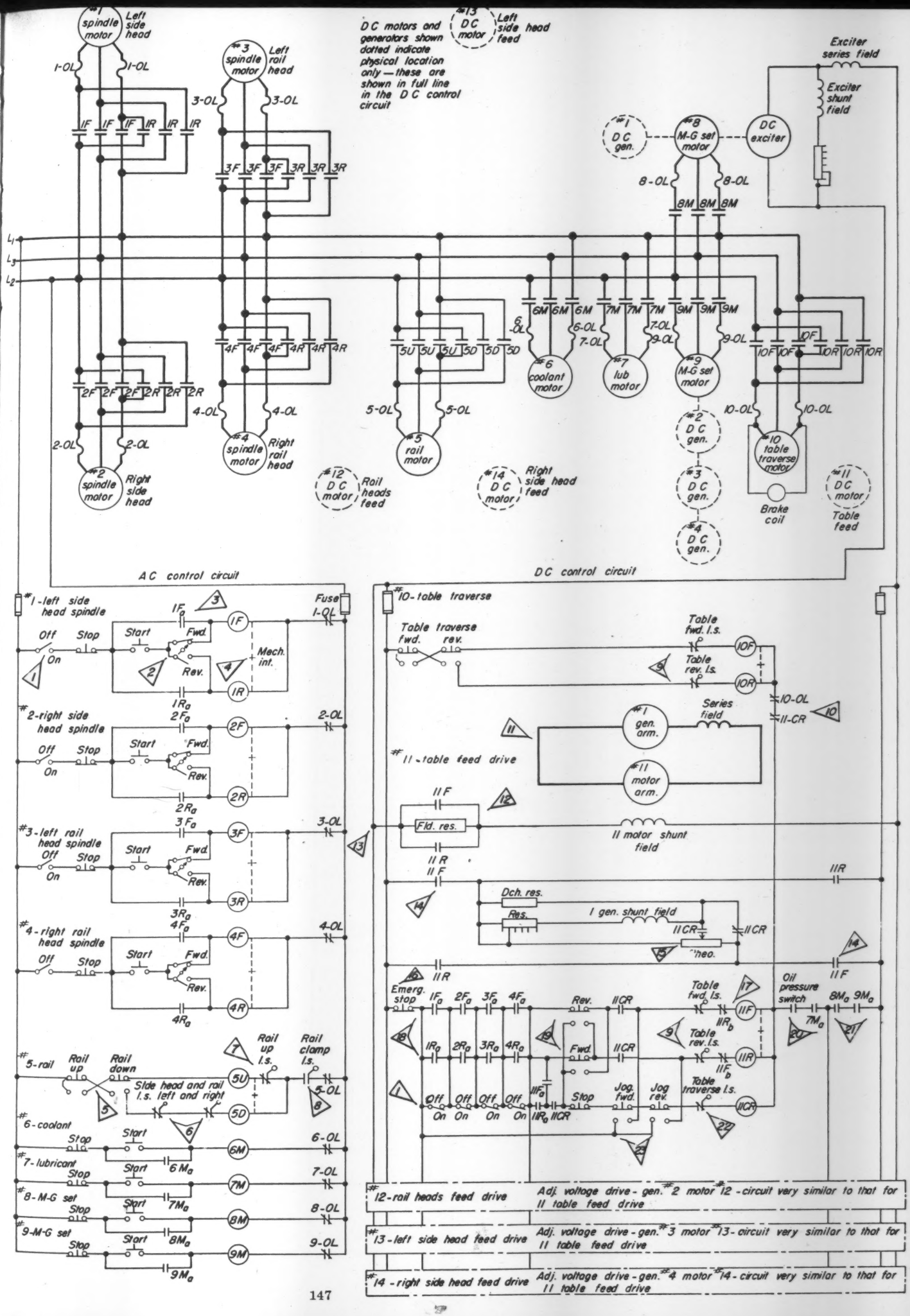
Movements of the left and right rail heads are in, out and tilt. The rail itself travels up and down. Both rail-head spindles move up and down and rotate. Movements of the side heads and their spindles are similar to those of the rail heads. Travel of the table is back and forth. Levers are provided so that the operator can select the desired motion of the milling heads. Electrical interlocking of the drives with limit switches prevents the moving elements from running into stationary parts of the machine or other moving members. Extensiveness of the electrical interlocking is indicated by the number of limit switches which are listed in the table on Page 148.

Simplified schematic diagram for all the electrical apparatus required to operate this milling machine is shown in *Fig.* 39. The head spindles, rail, coolant and lubricant pumps, m-g sets, and table (for rapid traverse motion) are driven by constant-speed a-c motors. The table, rail heads, and side heads have adjustable-voltage drives for a wide range in the feed speed.

The ABC's of interlocking are indicated on the diagram by pointers. These are identified by the following:

1. Operator can run any one or all of the head spindles by placing the manually operated selector switch in the on or off position. The other contact of this selector switch is in the holding circuit of the d-c table feed drive
2. The single-pole, double-throw selector switch makes the start button effective for energizing either the forward or reverse direction contactor coil only
3. The auxiliary contact on the directional contactor

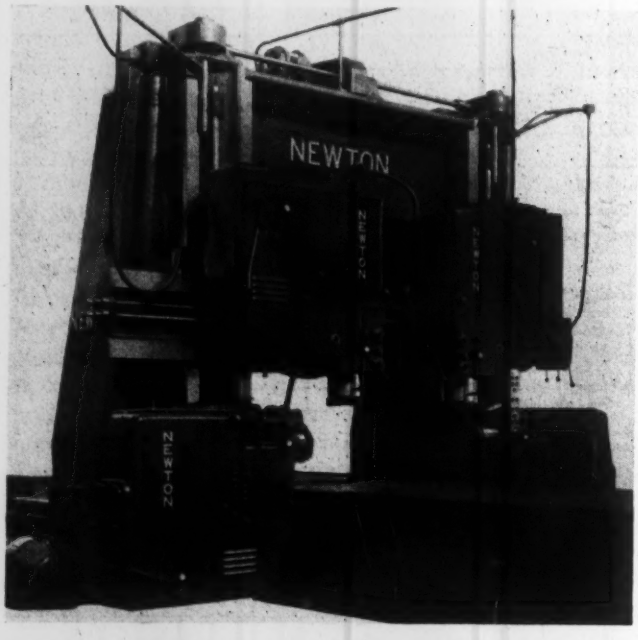
Fig. 39—Right—Simplified schematic diagram for milling machine. Interlocking features are identified by arrows


#1 spindle motor Left side head
#3 spindle motor Left rail head
#4 spindle motor Right rail head
#2 spindle motor Right side head
#13 DC motor Left side head feed
#12 DC motor Rail heads feed
#14 DC motor Right side head feed
DC motors and generators shown dotted indicate physical location only — these are shown in full line in the DC control circuit
#1 DC gen.
#8 M-G set motor
DC exciter
Exciter series field
Exciter shunt field
#2 DC gen.
#3 DC gen.
#4 DC gen.
#5 rail motor
#6 coolant motor
#7 lub motor
#9 M-G set motor
#10 table traverse motor
Brake coil
#11 DC motor Table feed
AC control circuit
DC control circuit
#1 - left side head spindle
#2 - right side head spindle
#3 - left rail head spindle
#4 - right rail head spindle
#5 - rail
#6 - coolant
#7 - lubricant
#8 - M-G set
#9 - M-G set
#10 - table traverse
#11 - table feed drive
#12 - rail heads feed drive
Adj. voltage drive - gen. #2 motor #12 - circuit very similar to that for #1 table feed drive
#13 - left side head feed drive
Adj. voltage drive - gen. #3 motor #13 - circuit very similar to that for #1 table feed drive
#14 - right side head feed drive
Adj. voltage drive - gen. #4 motor #14 - circuit very similar to that for #1 table feed drive

provides a holding circuit for overload and low-voltage protection

4. The mechanical interlock prevents more than one set of contacts from being closed at one time
5. The up and down buttons for the rail are electrically interlocked so that both coils cannot be energized at the same time
6. These limit switches prevent the rail from being driven down into the side heads
7. This limit switch prevents the rail from being driven past its upward limit
8. This limit switch prevents the rail from being moved in any direction unless the rail clamp is unfastened
9. These limit switches prevent the table from being driven past its forward and reverse limits
10. When the No. 11 table-feed drive is in operation, the 11*CR* relay prevents the No. 10 table-traverse drive, for rapid movement of the table, from being effective
11. For adjustable-voltage drives, the generator and motor armatures can be connected in parallel. No contactors are required
12. To prevent the motor from overheating at standstill, a "field economy resistor" is inserted in the circuit. When the motor is operating, the ventilation is sufficient to limit the temperature rise of the motor
13. Motor shunt field is solidly connected to the unfused d-c power supply. This method of connecting the field prevents field loss if the fuse should operate. Also, by having no contacts in series with the field, the hazard of no excitation current through the field is minimized
14. Motor direction is reversed by reversing the generator field. A change in the polarity of the voltage applied to the motor will reverse the direction of the motor armature as the motor field polarity is always the same
15. Generator field rheostat can be used to control the feed-motor speed
16. All four of the feed drives can be stopped by operating the emergency stop button
17. Normally-closed auxiliary contact on the opposite directional contactor provides electrical interlocking
18. None of the feed drives can be operated unless the head spindles are in operation. The operator can bypass the head-spindle interlock contacts with the selector-switch contact described in (1) if that spindle remains idle. If a spindle should stop, the feed will be immediately stopped. This prevents ramming the work into a stationary cutter
19. The forward and reverse pushbuttons are electrically interlocked in the 11*F* and 11*R* contactor coil circuits so that only one of the coils can be energized at the same instant
20. The table cannot be moved by either the rapid No. 10 traverse or No. 11 feed drive unless the No. 7 lubricant pump is in operation and the oil-pressure switch is closed. The ways of the table must be lubricated before the table is moved
21. None of the feed drives can be operated unless both of the m-g sets are in operation
22. The table-traverse limit switch renders the forward and reverse buttons ineffective when the lever is thrown to the traverse position. This is for rapidly moving to the desired position
23. The jog buttons can be used for positioning the work. The table moves only as long as the buttons are held down by the operator.

Fig. 40—Unit-head, planer-type milling machine utilizes twenty limit switch contacts, indicating the extent to which interlocks are employed to safe-guard machine operations

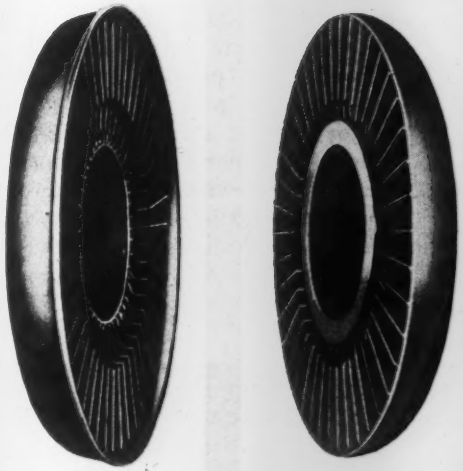


Limit Switch Interlocks for Milling Machine

No.	Limit Switches	Contact*	Operation
2	Table forward	N.C.	Opens at limit of table travel in forward direction
2	Table reverse	N.C.	Opens at limit of table travel in reverse direction
2	Rail head out	N.C.	Opens at limit of rail head travel in out direction
1	Rail head in	N.C.	Opens when rail heads approach each other while moving in
2	Side head down	N.C.	Opens at limit of side heads travel in down direction
2	Left side head and rail	N.C.	Opens when left side head and rail approach each other
2	Right side head and rail	N.C.	Opens when right side head and rail approach each other
1	Table-feed traverse lever	N.C.	Opens when table-feed traverse lever is thrown to traverse position
1	Rail-head feed traverse lever	N.C.	Opens when rail-head traverse lever is thrown to traverse position
1	Left side-head feed traverse lever	N.C.	Opens when left side-head-feed traverse lever is thrown to traverse position
1	Right side-head feed traverse lever	N.C.	Opens when right side-head-feed traverse lever is thrown to traverse position
2	Rail clamp	N.O.	Closes when rail is unclamped
1	Rail up	N.C.	Opens at limit of rail travel in up direction

*N.O. = Normally-open; N.C. = Normally-closed.

Fluid Coupling or Torque Converter?



What each type of drive has to offer is discussed in the following three related articles by authors intimately concerned with their development and application

Variable-Speed Fluid Drives

By John D. Rosebrough
Hydraulic Coupling Div.
American Blower Corp.
Detroit

OF THE two general types of fluid drives —the constant-speed and variable-speed—the former has achieved the most popularity through its use on automobiles. Consisting of an impeller, a runner, an enclosing casing, and an oil seal, it is filled with an initial charge of oil and sealed. This type is called a constant-speed unit because in normal operation the

output shaft runs constantly at 97 to 98 per cent of the input speed. The variable-speed fluid drive, on the other hand, does not use an oil seal, but has means of getting oil in and out of the rotating parts. A stationary scoop tube reaches inside the rotating parts and skims off fluid, circulating it outside where it can be controlled, the oil being returned by another passage.

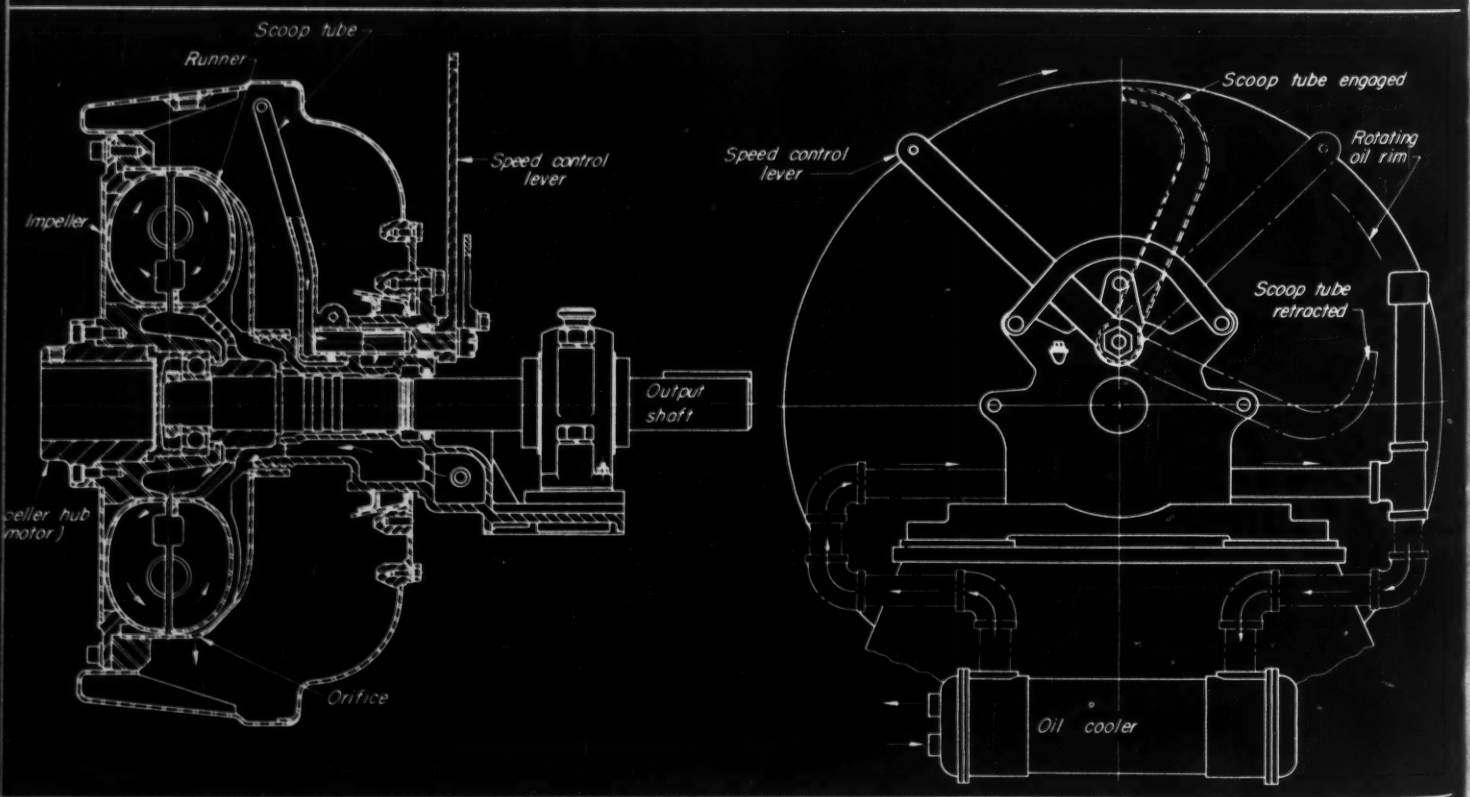
Looking at the unit in cross section, *Fig.* 1, it is seen how the fluid circulates from runner to impeller. The oil is pumped outward in the impeller, crosses the gap into the runner and flows inward. Actually, the fluid follows a spiral path, as it has motion in the direction of rotation also. A difference in speed between the impeller and the runner is required to maintain the oil circulation and to transmit the power. This difference, called slip, is 3 per cent or less under normal full load, the corresponding efficiency of power transmission being 96 to 97 per cent. The selection chart, *Fig.* 2, is based on this amount of slip.

CONSTANT-SPEED DRIVES: Primarily, the constant-speed unit acts as a centrifugal clutch and a shock absorber. It permits a motor to attain nearly full speed before picking up the load, greatly reducing motor current and simplifying the starting of heavy loads. A general-purpose motor with simple starter can be used where much more complicated starting equipment would otherwise be needed. On conveyors, crushers and similar loads, the shock absorption provided by the oil vortex extends the life of equipment as much as five to ten times and correspondingly reduces lost time due to emergency shutdowns.

VARIABLE-SPEED FLUID DRIVES find their greatest use in connecting a constant-speed electric motor to a machine being driven at varying speeds (a fan, a pump, etc.). They have some special application for use with engines and variable-speed motors but these are in the minority. Change of speed is accomplished by changing the quantity of oil in the oil vortex. When the circuit is full of oil the output shaft runs at 97 to 98 per cent of motor speed. When oil is withdrawn, the output shaft slows down to some lower speed. It is thus possible to adjust the output speed steplessly and accurately by gradually changing the fluid level in the rotors. The amount of oil in the vortex is adjusted in one of two ways: By a movable lever-controlled scoop tube or by a fixed scoop tube in conjunction with an auxiliary oil control pump.

Illustrated in *Fig.* 1 is a lever-controlled fluid drive, available in a range from one to 300 hp, the hub of which mounts on the motor shaft. Idle oil, not in the vortex between impeller and runner, is stored centrifugally in the rotating outer casing in an annular ring. To fill the vortex, the movable scoop is immersed in the oil ring; it then picks up the oil and forces it outside through a cooler and back into the

Fig. 1—Variable-speed fluid drive consists of fluid coupling with scoop tube which is used to change quantity of oil in circuit. Output-speed variation results when part of fluid is removed from the working circuit



working circuit. A continuous stream of oil leaves the vortex through a calibrated nozzle and enters the outer casing where it is again picked up by the scoop tube and circulated. Looking at the scoop tube in the end view, *Fig.* 1, it is seen how it is possible to vary the position of the tip of the tube and thus adjust the oil quantity to any desired level by moving the external speed-control lever.

Torque and Speed Relations

CHARACTERISTICS OF VARIABLE-SPEED FLUID DRIVE: When oil is withdrawn from the rotors of a fluid drive, its ability to transmit power is reduced and consequently the output shaft will slow down. By this means stepless speed variation can be achieved.

Starting Torque which this drive will transmit when the rotors are filled is equal to or greater than the peak torque of the motor normally selected for use with it. If the motor has a pull-out torque of 225 per cent, then 225 per cent torque is available through the fluid drive for starting the load. The fluid drive takes advantage of the "pull-out" torque of the motor to start the load and, as a result, the starting torque is much higher than that of the motor alone.

Range of Speed Control depends on the nature of the load. A smooth constant-torque load with considerable inertia, such as a paper slitter, can be driven continuously at any speed over a range of 3 to 1 (even slower for starting conditions). A fan or centrifugal pump may be driven as low as 15 to 20 per cent of full speed, although 3 to 1 is the normal operating range employed in most installations. A fluctuating torque load such as a vacuum filter or ball mill should not be operated continuously below half speed.

Speed Regulation Accuracy is also dependent on the nature of the load. In general, however, the speed regulation is the reverse of a direct-current motor—it is excellent at high speed where accuracy is most demanded and becomes progressively less accurate as the output shaft slows down. With mechanical draft fans and similar variable-torque loads, automatic controls have no difficulty in getting regulation to less than one per cent of the motor speed. With smooth rolling constant-torque loads, operators of machines generally find that the accuracy is well within the limits required for the process. With fluctuating torque loads, the fluid drive should not be considered if less than 2 per cent speed variation is required at top speed and 3 to 5 per cent at half speed. The percentage of fluctuation will depend on how much variation in torque occurs. It must be remembered that the natural cushioning ability of fluid drives cannot be secured without some "softness" of the drive and response to load change.

Overload Capacity is determined by the amount of oil in the coupling. With a full circuit, the oil vortex will transmit the peak torque of the motor. By gradually withdrawing (or adding) oil any degree of allowable overload can be preselected. It is possible, then, to limit output for such uses as tension control in winding, drawing tubes, and for extruders.

Incorporating Special Controls

AUTOMATIC AND REMOTE CONTROL: When remote or automatic control is used, either (a) a motor must be energized to move the speed control lever of the lever-controlled unit or (b) valves must be energized which admit oil to or release oil from the fluid drive to change speed of the pump-controlled unit, the control motor being actuated by any automatic control impulse which can make an electric contact. The automatic control may be measuring water pressure from a pump driven by the fluid drive, or air flow from a fan, or tension in steel strip being wound, or speed of a conveyor, or motor current on an agitator drive, etc. These are fairly common cases.

APPLICATIONS of fluid couplings to electric motor drives extend to every major industry. For easy

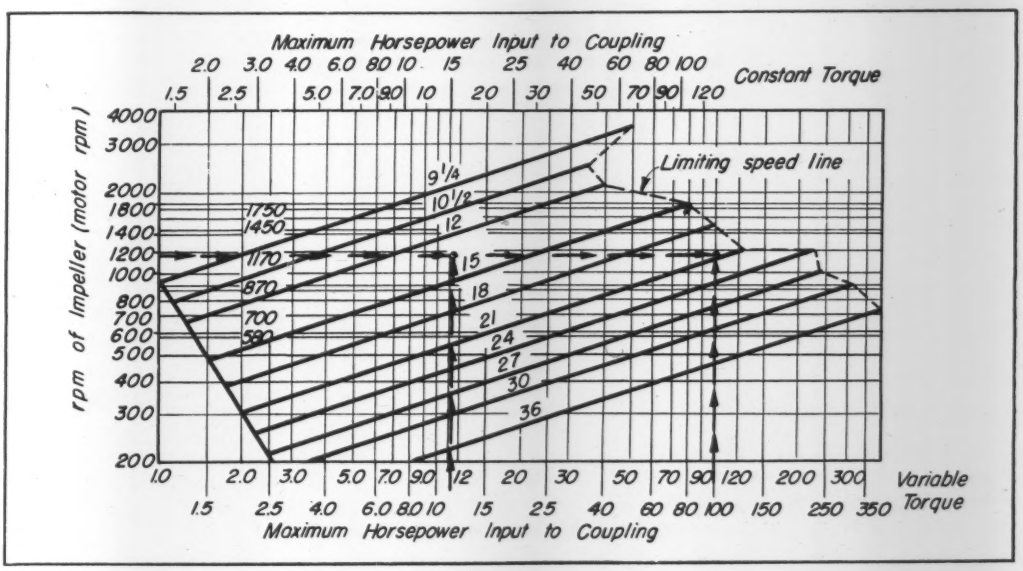


Fig. 2—Selection chart for fluid couplings shows sizes for various combinations of motor speed and horsepower. Horsepower varies as cube of input speed and as fifth power of the coupling diameter

analysis, the various applications are classified as follows for nature of load:

Variable-torque load: Fan or centrifugal pump. The power demand of these units varies as the cube of rpm under average conditions, and speed range can be 5 to 1 or more.

Constant-torque load: Horsepower varies directly with speed since friction is the load. These machines are further classified as: (1) Smooth rolling load with rolling momentum; printing presses, paper slitters, and triplex pumps are typical examples. A range of 3 to 1 is quite practical for continuous operation, and lower speeds can be held during the starting period. (2) Fluctuating torque loads; vacuum filters, crushers, rotary kilns and ball mills are typical examples. Two to one speed range is the maximum recommended; lower speeds can be held however during a short starting period by means of manual control.

Some of the machines which are using variable-speed fluid drives with considerable improvement in operation are textile spinning frames, paper coaters and slitters, agitators and mixers, ball mills, rotary kilns and dryers, drawbenches, extruders, cable and rope stranders, winches, and wire-drawing machines. The reasons for the use of fluid drive vary with the nature of the machine but, in general, one or more of the following gains is made:

(a) Use of sturdy constant-speed a-c motors for variable-speed drives

(b) Stepless variation of speed

(c) No-load starting of the motor regardless of the nature of the driven machine (simplifies motor starting equipment)

(d) A heavy inertia load can be started with a smaller motor without overheating the motor, since the motor runs at full speed where current is low and ventilation is good. Acceleration can be extended to reduce the power required

(e) Controlled gradual starting for machines which require it

(f) Positive overload limitation. There is no torque conversion at reduced speeds

(g) Shock absorption, extending the life of machinery. The vortex of oil is not a rigid coupling and cannot transmit sudden damaging overloads. Torsional vibrations are also absorbed

(h) Machine can be brought to a complete stop without stopping the motor by declutching the fluid drive

(i) Automatic control can easily be adapted to moving the speed control lever.

Three-Stage Torque Converters

By W. B. Gibson
Hydraulic Division
Twin Disc Clutch Co.
Racine, Wis.

TWO basic steps are required in the application of a hydraulic torque converter to a machine. The first is the correct application of the converter to the prime mover and its consideration as a power unit with a performance characteristic entirely different from that of the prime mover. The other is the application of the power unit and its performance characteristic to the complete machine. Since the power unit's performance characteristic is automatically infinitely variable within limits, and since each point on the performance curve is related to output efficiency of the unit, the performance requirements of the machine as a whole must be more closely considered than is generally true when applying gear transmissions.

In this article, discussion will be confined to applications of the Lysholm-Smith three-stage hydraulic torque converter, of which over 100,000 successful applications have been made in this country to prime movers ranging from 50 hp to 750 hp in machines for almost every type of commercial use. The design comprises a closed-circuit hydraulic unit *Fig.* 3, consisting of a centrifugal pump or impeller driven by the engine or prime mover, a three-stage turbine

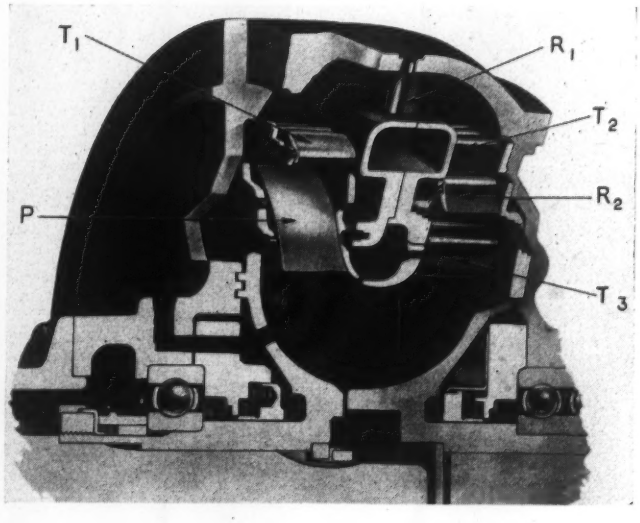


Fig. 3—Right—Cutaway view of three-stage torque converter shows details of centrifugal pump P, three turbine stages T_1, T_2 and T_3, and two reaction stages R_1 and R_2

attached to and driving the output shaft, and two sets of stationary or guide blades attached to the stationary housing. The only restriction to the flow of the working fluid discharged by the centrifugal pump into the turbine circuit is that offered by the several sets of blading, the varying rotational velocity of the turbine, and attendant fluid friction losses. In *Fig.* 3, P is the centrifugal pump, T_1, T_2, and T_3 are the three turbine stages and R_1 and R_2 are the two reaction stages. Horsepower capacity can be accurately controlled by the fifth-power law of the pump diameter and varied by the cube law of input rpm.

Accurate comparative ratings of converters require that a certain basis be established, such as a constant input rpm to the centrifugal pump and varying rpm on the turbine from full racing to complete stall. Since the turbine rpm is controlled solely by the load imposed on it, this load is also reflected back to the prime mover, tending to pull it down in speed. Therefore, to maintain constant input speed more torque or horsepower is required. This determines the "primary" torque or horsepower curve for a given converter. With a constant input rpm and varying output rpm, the output rpm expressed as a

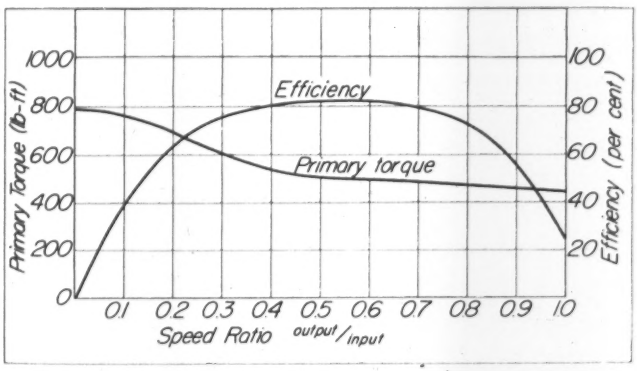


Fig. 4—Primary torque and efficiency curves for three-stage hydraulic torque converter. Primary torque is the input torque for constant input speed

decimal fraction of the input rpm is known as speed ratio. At the time primary torque or horsepower capacity is determined for each speed ratio factor, the unit efficiency is also determined. Such a primary curve is shown in *Fig.* 4. From this it can be seen that for any given speed ratio there is a fixed value of input torque or horsepower required and the converter has a fixed efficiency. Most primary curves are shown as torque curves.

It is not impractical to maintain constant input rpm and horsepower, but since the relation of speed ratio and efficiency remains constant for all practical purposes these values may be converted to primary torque by the following variation of the horsepower formula:

$$T = \frac{Hp \times 5250}{Engine\ rpm} \times \left(\frac{Rating\ rpm}{Engine\ rpm}\right)^2$$

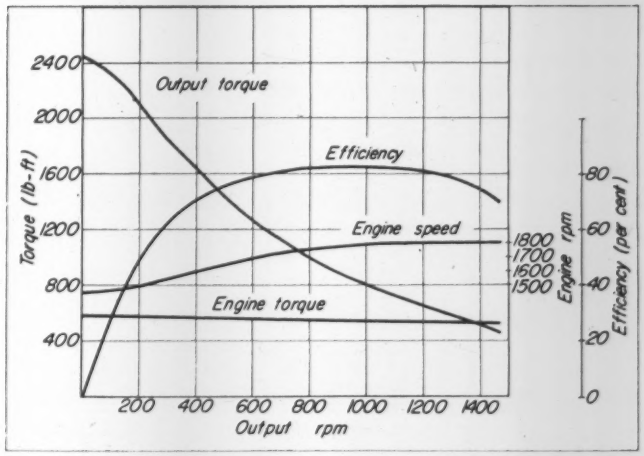


Fig. 5—Characteristics of a power unit consisting of an internal combustion engine and three-stage converter

By proceeding in this manner for all points of a fixed-throttle speed range of the prime mover (usually full throttle) the output torque and speed characteristics of the engine-converter power unit can be determined. These are usually shown as in *Fig.* 5.

Such methods, although simple once understood, tend to throw a cloud of mystery around the converter and its application. As stated before, the speed ratio-efficiency relation remains practically constant for varying input rpm and horsepower, and for this reason the primary torque curve for a given converter can be expressed in the form shown in *Fig.* 6. Here the speed ratio—efficiency relation is expressed as a family of curves showing the converter horsepower capacity at the various speed ratios when the input speed is known. Obviously the output speed is the input speed times the speed ratio and output horsepower is input horsepower times efficiency. Plotted as output horsepower against output speed this curve will take the form of the efficiency curve shown in *Figs.* 4 and 5. Plotted as output torque against output speed the curve takes the form of the output torque curve shown in *Fig.* 5. This torque converter is concerned only with the true horsepower at its input shaft and therefore all parasite loads must be deducted from the engine curve before making an application.

Factors in Selecting Torque Converter

CHARACTERISTICS: The following four converter characteristics must be considered when making applications: Utility ratio, overall torque ratio, stall torque ratio, and efficiency. Of these, efficiency is of primary interest when applying the converter to the prime mover while the remaining three characteristics are of more concern when applying the power unit to the complete machine.

Efficiency: It will be noted from *Fig.* 6 that converter efficiency is zero at both stall and racing

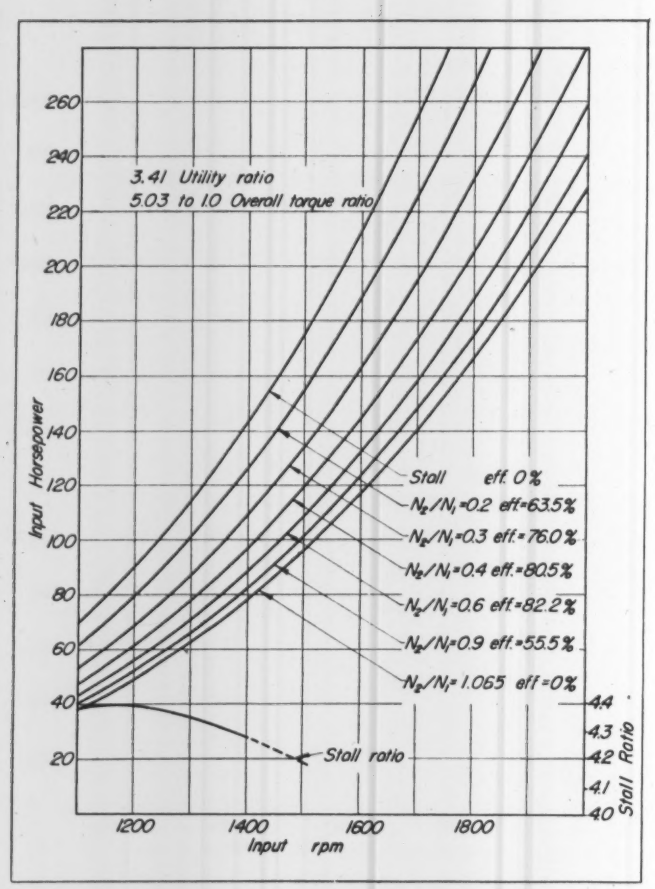


Fig. 6--Horsepower capacity of a typical three-stage converter at various speed ratios

points, reaching a maximum of 82.5 per cent at 0.55 speed ratio. Since cooling is provided for only 30 per cent of input capacity, any continuous operation of the unit under full power at speed ratios where the efficiency is less than 70 per cent will cause overheating. Further, this is impractical except in very special applications, known beforehand, where adequate cooling is provided. Therefore, normally, the converter is matched to the engine so that the full power at governed engine speed is absorbed by the converter at a high speed-ratio point where the converter efficiency is between 70 and 80 per cent.

Application of the power unit to the machine should be such that the normal, light, running loads will demand output torque from the converter of sufficient magnitude to limit continuous operation at the high speed-ratio of 70 per cent efficiency or better just mentioned. Short periods of high-speed, low-efficiency operation are not detrimental.

Utility Ratio: Similarly, continuous operation in the high-torque low-speed range with efficiencies below 70 per cent will cause overheating. Primarily, this range is for the starting and rapid acceleration of heavy loads or for emergency use where above-normal torque requirements exist. Therefore, the normal continuous-duty operation of the machine should be confined within those two speed ratio points representing 70 per cent efficiency. The high speed-ratio factor at 70 per cent efficiency divided by the low speed-ratio factor at 70 per cent efficiency is the utility ratio and represents the efficient operating range of the converter. Frequent short excursions outside this range, spaced by ample cooling periods of operation within this range, are normal practice.

Overall Torque Ratio: Referring to *Fig.* 5, the output torque at the high-speed 70 per cent efficiency point divided into the output torque at stall gives the overall torque ratio or torque range throughout which the power unit is normally expected to operate. The value should be in the order of 5 to 1.

Stall torque ratio is the output torque at stall divided by the input torque at stall and should range from 4 to 1 up to 6 to 1. This factor applied to the prime-mover torque shows the torque available from the power unit to get a load off center or started, and is generally shown as a curve as in *Fig.* 6 since the factor will vary slightly with input rpm.

The designer who is considering the application of a hydraulic torque converter must consider all four factors as applied to the work requirements of his machine or its duty cycle. Contrary to general belief, the order of their importance is as listed in the foregoing where converter efficiency is listed last. While the hydraulic torque converter is not the answer to all power transmission problems, this is not due to its efficiency characteristic, as often believed. Other characteristics show up in the ability to produce more work in a given time with greater ease and less maintenance and overall operating costs than the conventional mechanical drive. This sets the overall or operational efficiency of the complete machine. Therefore, in making a converter application the first three characteristics should be used to the best advantage with respect to the work requirements of the machine. This done, converter efficiency alone becomes less important.

Controlling Performance Characteristics

The foregoing four characteristics are closely interrelated and generally the improvement of one is at the expense of the others. They can, however, be controlled and changed to suit the work requirements of specific machines. For example, in free-rolling vehicles such as buses and railcars where the normal work is at high speed and low torque but rapid acceleration is desired, a high stall-torque factor is the important characteristic and is desirable even with a sacrifice in the other three. Here the peak converter efficiency moves to the left or over to the high-torque low-speed range of the converter. Since the converter is by-passed for the high-speed operation, its performance at the high speed end is not so important.

Crawler-type tractors generally require the peak converter efficiency to occur in the high- speed ratio range since they normally operate at a fixed throttle setting and are designed to operate within well de-

fined vehicle speeds. This also can be accomplished by slight compromising of the other characteristics. Proper gearing of the converter output shaft to the tracks does the rest.

Characteristics of a converter favoring a middle of the road course appear in *Fig.* 4 where the efficiency curve is quite broad, providing a 3.41 utility ratio. The stall torque ratio peaks at 4.4 to 1, providing an overall torque ratio of 5.03 to 1.

Many machines require a wider operating torque and speed range than can be provided by the torque converter alone. Generally, two speeds behind the converter suffice to cover the full required range but additional ratios can be used. The spread between ratios is much greater than with conventional gear transmissions, ranging from 2.5 to 1 up to 2.8 to 1 because of the torque-speed range of the converter. Any reversing requirement is provided mechanically to suit the application. The ultimate goal is a completely integrated power-operated transmission.

Single-Stage Torque Converters

By R. Brunken and W. Howard
Torcon Corporation
Chagrin Falls, O.

COMBINING a hydraulic torque converter (Schneider system) and a hydraulic coupling into a single unit, the assembly shown in *Fig.* 7 retains the advantages of each. Transition from the converter range to the coupling range is effected automatically by the inherent characteristics of the unit, no outside controls being required. This combination makes it possible to operate at high efficiencies over a wide speed range and to obtain high output-shaft speeds, permitting full utilization of the engine horsepower throughout the entire range of part throttle to full throttle operations. These outstanding features make the unit, which is called a Torcon, an ideal automatic transmission for such equipment as logging machinery, oil well machinery, locomotives, rail cars, cranes, shovels, and road machinery.

A cross-sectional view is shown in *Fig.* 7. The main functional parts are the converter pump, the converter turbine and the reaction member. These are single-piece aluminum castings, comprising vanes or blades which direct the flow of oil in the hydro-kinetic system.

Engine power is transmitted from the flywheel through a gear type coupling to an input shaft and then through the pump cover to the converter pump. The turbine is mounted opposite the converter pump and, by means of a splined hub, is attached to the output shaft. The reaction member is mounted between the converter pump and turbine. It receives the oil after it leaves the turbine and redirects it into the pump. The function of the reaction member is to increase torque in the converter range. In the coupling range no torque increase is desired and therefore the reaction member is arranged to be inoperative during this phase. This effect is attained by mounting the reaction member on the stationary sleeve by means of a freewheeling unit. This permits rotation of the reaction member during the coupling range and prevents rotation during the torque converter range. Whether or not

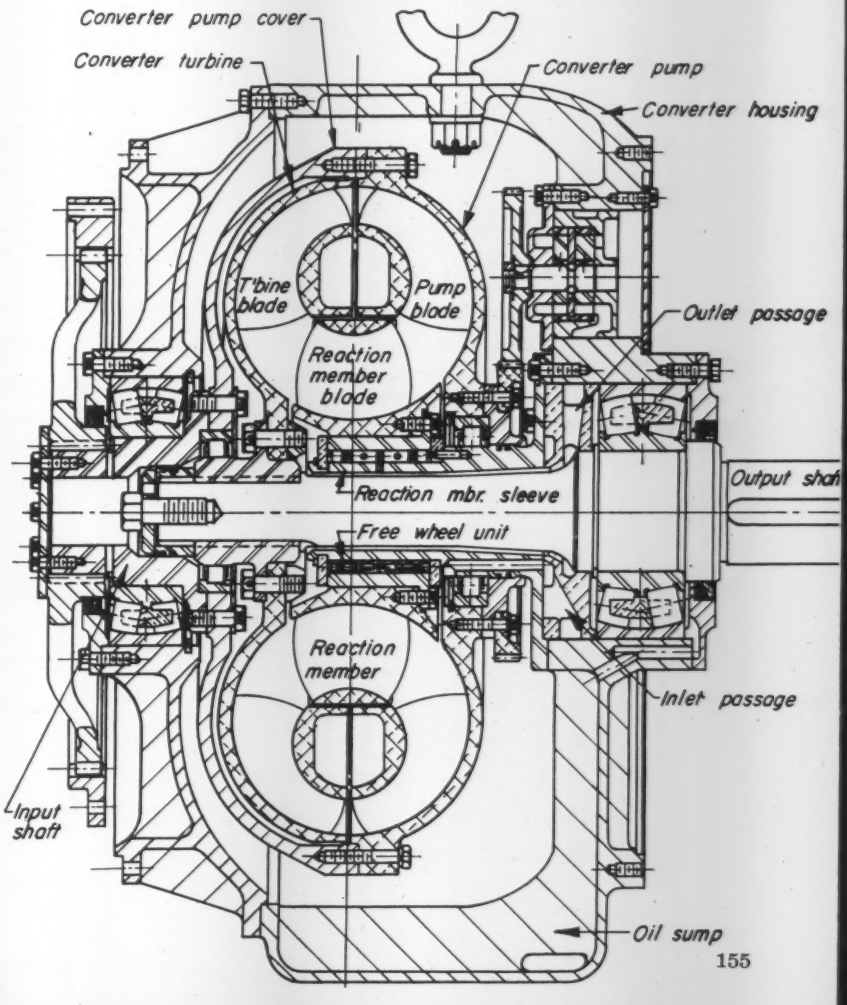


Fig. 7—Cross section through single-stage torque converter in which reaction element freewheels when torque ratio becomes unity. Converter then acts as fluid coupling

the reaction member rotates depends upon the direction in which the oil leaving the turbine impinges against its blades. The freewheeling unit is of the sprag type, concentricity of the inner and outer race being assured by means of roller bearings on either side of the sprags.

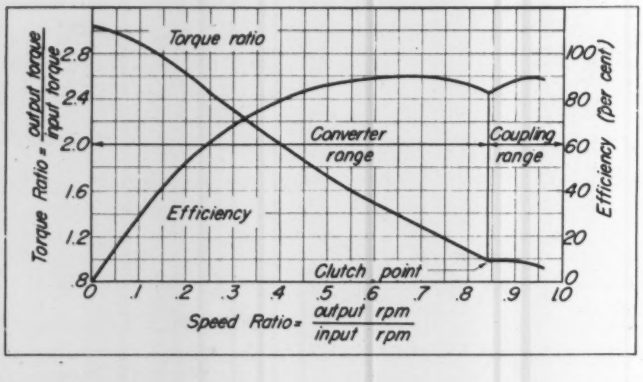
In addition to the main functional parts, the unit is equipped with an oil pump system comprising two pumps in one housing. The forward or pressure pump draws oil from an oil tank and forces the oil through the inlet passage into the converter proper. This oil also serves to lubricate the bearings. After circulating through the converter the oil passes through the space between the reaction member sleeve and the output shaft and then through the outlet passage to a pressure relief valve. It then flows through a heat exchanger and back to the oil tank.

Leakage Oil Acts as Lubricant

The oil is retained by simple piston ring type oil seals. These seals permit the leakage of a small amount of oil which is used for lubricating the front spherical roller bearing and the oil pump drive gears. The oil eventually accumulates in the oil sump from where it is picked up by the scavenger pump and returned to the oil tank. Two external seals of standard lip type are used to seal the input and output shafts. The bearings have been designed so that, if desired, the output drive may be by means of chain and sprocket without the necessity of an outboard bearing.

A general idea of the performance of the Torcon unit can be gained from *Fig.* 8. The curves are plotted against speed ratio and hold true for all operating conditions, both full and part throttle. At zero speed ratio the torque multiplication is highest, the standard multiplication being 3:1. This gradually decreases with increasing speed ratio until a point is reached where the output torque equals the input torque, a torque ratio of 1:1. The ratio of unity then prevails from this output speed on up to almost maximum engine speed. The efficiency curve increases rapidly as the speed ratio increases, reaching a maximum for the converter range at about 0.65 speed ratio, and then begins to drop again until a point is reached

Fig. 8—Efficiency and torque-ratio curves for the single-stage converter shown in Fig. 7



where the torque ratio is 1:1 as the coupling range begins. From here on the efficiency again begins to rise and continues to do so throughout the coupling range.

Typical performance curves when applied to an engine are shown in *Fig.* 9, where the characteristics show up to even better advantage. During the converter range the engine performs at a practically constant speed, making it impossible to stall the engine due to overload. After passing into the coupling range, the engine speed increases in nearly direct proportion to the output speed. The efficiency is comparatively high over a wide speed range, attaining 70 per cent at 400 rpm output speed and remaining above this value for the balance of the speed range at full throttle. This is due to the combined functional advantages of the torque converter and coupling.

The dotted curves in this figure indicate the operating characteristics at half throttle. It will be noted that here the performance is very similar ex-

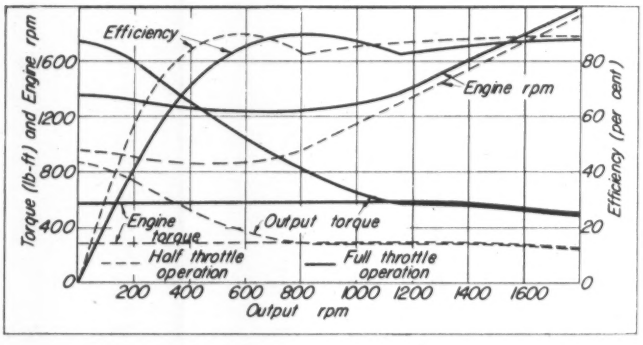
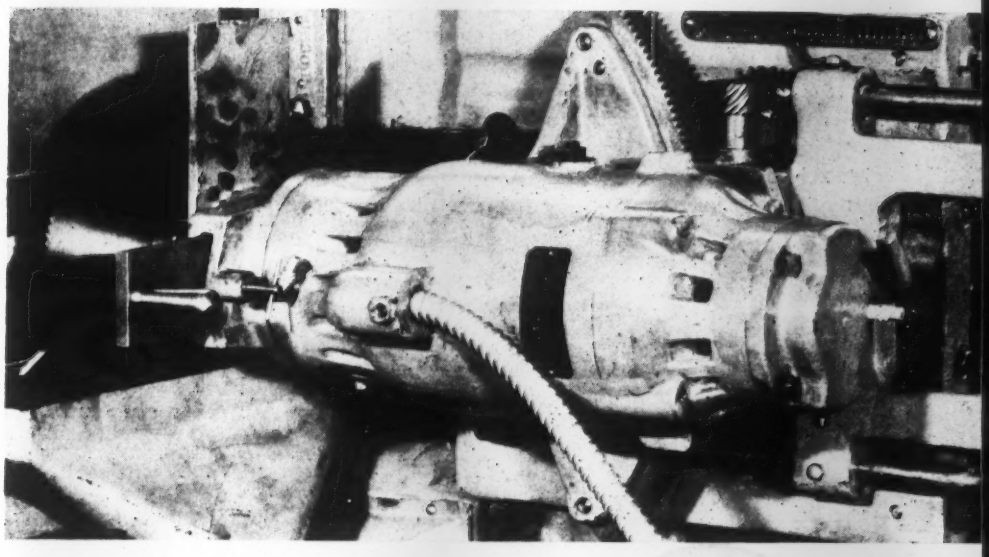


Fig. 9—Performance of a power unit consisting of an internal combustion engine and the converter in Fig. 7

cept that the converter range has been reduced and the coupling range correspondingly increased. This extends the high efficiencies over an even wider speed range. In this case all operation above 270 rpm will be at efficiencies higher than 70 per cent. Other part-throttle settings will give rise to similar curves.

Selection of the proper-size converter for a particular engine depends upon the application for which the converter is to be used and the desired operating speed of the engine itself. If it is permissible and desirable that the engine speed be close to governed speed during the whole operating cycle, the maximum performance is obtained by using a relatively small converter. During full-throttle operation, then, the unit will perform most of the time in the converter range. If, on the other hand, it is not desirable to utilize the full engine speed and horsepower during the converter range, a larger unit should be used. The full horsepower and speed will still be available during an extended coupling range. Other characteristics may be obtained by changing the blade contours, which will affect the maximum torque ratio, the efficiency and the power absorption capacity.



When You Design-In Shell Type Motors

By R. E. Arnold
Small and Medium Motor Division
General Electric Company
Lynn, Mass.

WHEN the design engineer purchases a complete motor he obtains not only a full set of electrical characteristics, but also a complete ventilation system and a mechanical system which includes bearings, lubrication, dynamic balance, assembly, mounting, and accessibility. While a shell type motor (*Fig.* 1) has, of course, a set of electrical characteris-

tics, it has only a partial ventilation system and never has a complete mechanical system. Therefore, the price the machine designer must pay for the compactness and flexibility of the shell type motor is the assumption of responsibility for some of the functions that are those of the motor manufacturer on complete motors.

The greatest advantage of shell-type motors is that, being simple electromagnetic parts, they can be easily adapted to any environment. A second advantage is that at least three, and sometimes six or seven horsepower ratings at a given speed are built in each of

TABLE I

Maximum Safe Speed for Laminated Silicon-Steel Motor Rotors

Diam. of Rotor (inches)	Revolutions per Minute
1	80,000
2	40,000
3	27,000
4	20,000
5	16,000
6	13,000
8	10,000

the seven principal shell diameters. Large horsepowers are available in small diameters because there are several lengths of stacking, longer than the customary designs, available in shell-type motors. However, special treatment of the ventilating system and bearings usually is required. Motors with 8, 10 and 12⅜-in. OD have been standardized by various manufacturers and 3.648, 4.703, 5.703 and 6.468-in. diameters have been standardized by common usage over the years.

Simplicity of the shell-type motor makes it readily adaptable to the solution of two broad classifications of special design problems: (1) Those where limitations of space exist or peculiar shapes are desired and (2) those involving unusual mechanical problems such as high speed, dynamic balance, critical speeds, or bearing design.

SPACE LIMITATION: Frequently not enough room is available to mount a motor of standard construction. The standard motor as designed by a motor manufacturer has flexibility, and by slight modifications must be able to meet a wide variety of application demands such as multispeed, slow speed, gearmotors, wound rotor, oversized bearings, etc. The mechanical construction and ventilation system must be large enough to permit such variations. For example, the motor ball bearings are designed to give ample life with a flat belt or V-belt drive, whereas with direct drive, smaller bearings can be used. Thus, where compactness is necessary, a small-diameter, long-length shell motor can be used advantageously with tailor-made bearings and ventilation schemes.

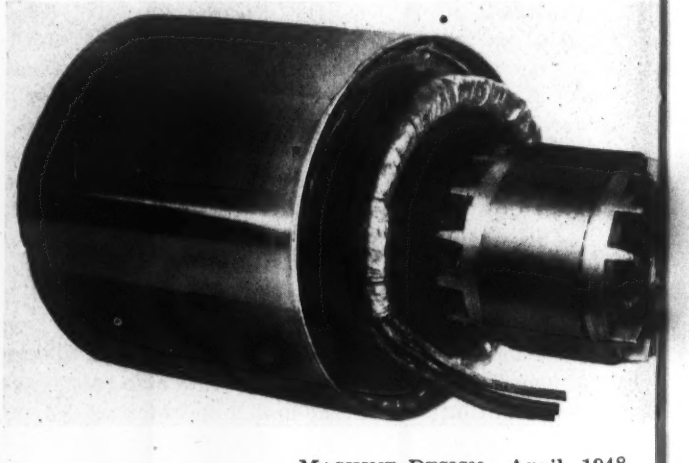
Motor designers have long related horsepower output of a motor to its electromagnetic volume, which is directly proportional to the length and the diameter squared of the punchings. Horsepower can be increased by increasing the length of the silicon-steel stacking, but this imposes ventilation problems because fan diameters are not increased and the heating problems may become acute because of the lack of ventilating air. Bearing problems are also introduced if the shell motor is too long, because it becomes difficult to maintain proper bearing size. However, small diameters are important when motors must be mounted closely to a grinding wheel or cutting tool because the effective working diameter of such tools is the amount the wheel diameter extends beyond the motor diameter (see photograph on previous page). Close spindle-center locations on direct-drive gang drills, precision borers, woodworking molding machines, and tenoners are other applications where the small-diameter shell-type motor is particularly advantageous. *Figs.* 2 and 3 portray excellent applications of shell-type motors for close-spaced spindle design.

HIGH-SPEED APPLICATIONS: Of course, 14,400 rpm of a 2-inch diameter rotor is no worse than 3600 rpm of an 8-inch diameter rotor. High speed should be defined in peripheral velocity starting at about 12,000 fpm. Velocities are being run as high as 25,000 to 30,000 fpm. TABLE I shows the approximate speeds at which laminated silicon-steel rotors can be run without unusual mechanical design to prevent failure of the punchings due to centrifugal force.

Higher than 60-cycle frequencies are necessary to obtain these high speeds. Such frequencies have been supplied by induction frequency changers, generators, and inductor frequency changers. The last mentioned has appeared on the market only recently. It has met with some favor because there are no copper wire windings on the rotor and it has a rugged, simple construction.

Extremely high speeds are required for internal grinding, and frequencies have gone as high as 2,000 cycles with very small motors running as high as 120,000 rpm. Woodworking machinery cutters operate at 7200 to 10,800 rpm in many cases; drilling and milling applications sometimes go to 14,000 rpm; and boring as high as 25,000 rpm. *Figs.* 4 and 5

Fig. 1—Below—Typical shell-type motor, as supplied, comprises stator (left) and rotor (right)



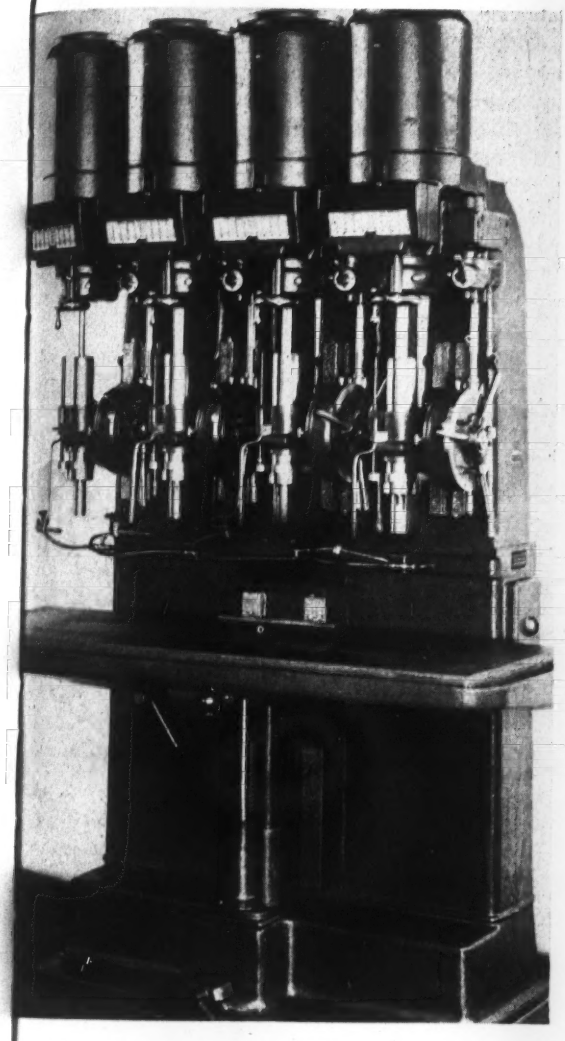


Fig. 2—Built-in shell-type motors permit close spacing of the spindles on this four-spindle vertical drilling machine

show high-speed grinder applications. When these speeds were reached a whole field of mechanical problems was uncovered. Critical speeds, bearings and lubrication, vibration and mechanical fits, ventilation and heating are some of the problems to be considered.

Electrical output, without any other considerations, will increase at least directly with the frequency for a given D^2L (diameter squared × length) of electromagnetic material. The available horsepowers at high frequencies are great and the savings in space and weight are particularly advantageous compared to 60-cycle motors.

As in all phases of engineering, advantages must be weighed against disadvantages in reaching practical conclusions. A disadvantage of the shell-type motor is its higher initial cost which is due to the fact that the machinery manufacturer must build and assemble a number of the motor parts. Then too, the shell-type motor generally has only slight resale value because it is a special. Another disadvantage, previously mentioned, lies in the fact that engineering responsibility is transferred from motor producer to machinery manufacturer in such matters as ventilation, bearings, lubrication, dynamic balance, assembly and mounting.

HEATING AND VENTILATION: The National Electrical Manufacturers Association Motor and Generator Standards (MG12-202) states, "The temperature rise of the shell-type motor depends on the design of the ventilating system as well as on the motor losses. The motor manufacturer's responsibility is limited to (a) supplying motors with losses, characteristics, current densities, and flux densities consistent with complete motors of similar ratings, size, and proportion, and (b) when requested, supplying information regarding the design of a ventilation system which will dissipate the losses within the rated temperature rise. Therefore, obviously the machine manufacturer is ultimately responsible for the temperature rise."

Losses: Losses must be dissipated to keep motor temperature rises normal and insure normal life of the insulation. Under steady-state running conditions, losses consist of friction and windage, core losses, stator and rotor resistance losses, and

TABLE II

Suggested Values for Minimum "Effective Areas" of Motor Ventilation Paths*

Motor Diam. (Inches)	Type of Ventilation**	Horsepower Ratings									
For 2-Pole, 36000-Rpm Motor		¼	⅓	½	¾	1	1½	2	3	5	7½
		Effective Areas (sq. in.)									
3.640	SERF	¾	1½	2½	3½						
	SEAF	½	1	1½	2	3½	5				
	DERF	½	¾	1¼	1¾	2½	4				
	DEAF				1¼	1¾	2½				
4.703	SERF			1¼	2½	4					
	SEAF			1	1½	2¼	3½	5			
	DERF			¾	1¼	1¾	3	4	6		
	DEAF					1¼	2	2½	4	6	
5.703	SERF					2	4	7			
	SEAF					1½	2½	4	6		
	DERF					1¼	2	3	4½	8	
	DEAF							2	3	5	8
6.468	SERF					1½	3½	6			
	SEAF					1¼	2¼	3	5	9	
	DERF					1	1¾	2½	4	7	
	DEAF							1½	2½	4½	7

	Horsepower Ratings									
For 2-Pole, 3600-Rpm Motor	1½	2	3	5	7½	10	15	25	30	40
For 4-Pole, 1800-Rpm Motor	1	1½	2	3	5	7½	10	15	20	25
For 6-Pole, 1700-Rpm Motor	¾	1	1½	2	3	5	7½	10	..	15
For 8-Pole, 900-Rpm Motor	½	¾	1	1½	2	3	5	7½	10	15
	Effective Areas (sq. in.)									
8.000 SERF	3	5	9							
SEAF	2	2½	4½	8						
DERF	1½	2	3½	6	10					
DEAF				3½	6	10	13	20		
10.000 SERF				9	15					
SEAF				5	8	12				
DERF				3½	6	9	15			
DEAF						6	10	16	21	30
12.375 SEAF						8	12	20		
DERF						6	10	15	25	30
DEAF								10	14	20

** SERF—Single-end rotor fan. SEAF—Single-end auxiliary fan. DERF—Double-end rotor fan. DEAF—Double-end auxiliary fans.

* Values are for straight-blade fans. For single-direction meters, refinements such as curved-blade fans, scrolls and diffusers can be used.

Note: Diameter of auxiliary fan, if used, is assumed to be 8/10 of the motor shell diameter for 2-pole motors above 50 cycles, and equal to the shell diameter for all others. "Effective area" should be changed inversely proportional to fan diameter.

Effective area should be exceeded as much as possible in order to provide a margin of safety. Increase the suggested minimum value 50 to 100 per cent for motors used on special duty cycles; for high-slip motors 8 to 13 per cent; for 50-cycle motors 25 per cent.

For 25-cycle motors read "effective area" from the tabulation for a fictitious hp rating equal to 3 times the real rating. Example: 5 hp, 25 cycles use area tabulated for 15-hp motor of the same diameter and number of poles.

For high-speed motors on high-frequency circuits, read the "effective area" from the tabulation for a fictitious hp rating equal to (60/frequency) x (hp rating).

For short-time rated motors, (1 hr, 30 min., 15 min., etc.) use the same "effective areas" that apply to continuous rated motors.

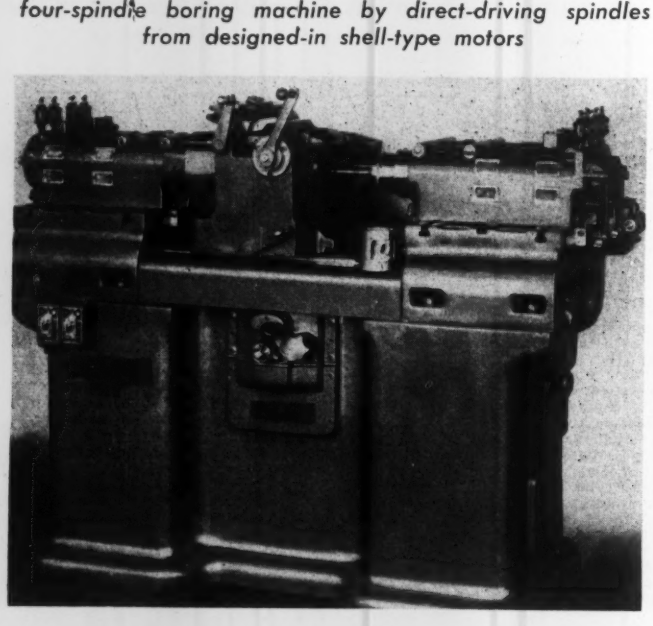
stray load losses. Values of these losses can be obtained from the motor manufacturer. Speed transients also cause losses. An acceleration from rest to full speed causes losses in the rotor winding equal to the kinetic energy stored. This loss equals

$$2.3 \times 10^{-4} \times \text{rpm}^2 \times WK^2$$

in watt seconds, where WK^2 is the moment of inertia, lb-ft^2, of the motor rotor and the external load at motor shaft speed. A stator loss of about 0.4 to 1.2 times as much as the rotor loss is associated with the rotor loss, depending upon the size of motor and type of design. For a plug stop, these rotor and stator losses are three times that of an acceleration, and for a reversal, four times that of an acceleration. The losses of dynamic braking are considered equal to an acceleration. Every jog is considered roughly equal to about one-third of an acceleration. Stalling adds large losses in both the stator and rotor. The loss in the rotor equals $N_sT/7.04$, where N_s is synchronous speed (rpm), and T is torque (lb-ft). The associated stator loss varies from 0.3 to 1.2 times as much depending upon the size of the motor and type of design. Losses from speed transients and stalling may be high and should be carefully reviewed.

Cooling: Dissipation of losses depends upon the enclosure of the motor and the type of cooling system used. Consider first totally enclosed non-ventilated construction. Totally enclosed motors are becoming more popular every day, particularly in high-speed, shell-type applications. This is due to the exacting vibration requirements and the fact that metal dusts deposit on fans and upset the dynamic balance of the apparatus. Also, cast iron dusts and poor atmospheric conditions can cause breakdown of open, drip-proof, and splash-proof motors. Losses are dissipated from totally enclosed motors by conduction and radiation. The conduction of losses is through the mounting, and the radiation of losses depends upon the external area of the motor housing, which is the radiation surface, and the surrounding material. It is hard to put a quantitative figure on the heat that is conducted through the mounting because the mounting arrangement varies with every application. However, 0.4 to 0.625 watts per square inch of radiation surface can be dissipated, depending upon the speed of the motor. Temperature rise can be determined approximately from this value. Some designers use ribbed construction to increase the radiation surface.

Fig. 3—Close spindle spacing was made possible on this four-spindle boring machine by direct-driving spindles from designed-in shell-type motors



Long-Motor Cooling Difficult

Long motors cannot be cooled easily because the horsepower output and the corresponding losses increase much more rapidly than the radiation surface. For example, assume a 2-hp rating at 1800 rpm, in a case about 10 inches in diameter and 8 inches long, with a 2-inch height of iron that is totally enclosed, non-ventilated, with a 55-C rise. If the height of iron were increased to 5 inches, the length of the case would be increased to 11 inches. The power could be increased to 7½ hp but the dissipation area would be increased only 24 per cent. It readily can be seen that even a 3-hp rating cannot be obtained unless the efficiency is materially increased.

Horsepower can be increased with frequency but the dissipation of the losses is not increased greatly with speed. Thus from 1800 to 3600 rpm only about 10 per cent more watts can be dissipated. Therefore, high output with high frequency cannot be obtained in the totally enclosed, nonventilated type of motor construction.

Water Cooling: Water or liquid cooling is the most effective means of cooling a motor. It takes only about one gallon of water per minute to cool a 25-hp motor with an efficiency of 85 per cent and 10 C rise of water temperature. Of course, if the water used is contained in a closed system an adequate reservoir must be maintained. With adequate water cooling, full advantage of long motors and high-frequency high speed can be realized. Outputs can be obtained at least directly proportional to frequency and somewhat more than proportional to length. Thus, with the totally enclosed, nonventilated motor of the previous example, if frequency were increased from 60 to 180 cycles, 20 hp could be obtained from the same electromagnetic parts.

OPEN MOTORS: Motors of open construction should have ventilation systems designed for particular applications. The key to the design of a good ventilation system is numerical calculation. Calculations can be simple, and complicated computations of temperature differentials and airflows can be circumvented by a simple numerical method which experience has shown to be practical and effective.

The drawings of *Fig.* 6 show some suggested schemes for cooling open motors, as well as four basic ways of cooling, namely, single-end rotor fans (*g*), single and auxiliary fan (*j*), double-end rotor

fans (*a,b,c,d,e,* and *h*), and double-end auxiliary rotor fans (*f* and *i*). Select a general design most appropriate for a particular installation, make a sketch (preferably to scale) of the assemblage of parts which forms the air passages, determine the area of restrictions, and calculate the effective area by the formula:

$$\text{Effective Area} = \sqrt{\frac{1}{\frac{1}{A_a^2} + \frac{1}{A_b^2} + \frac{1}{A_c^2} + etc.}}$$

where A_a, A_b, and A_c are the areas in square inches of the restrictions. Then refer to TABLE II for the minimum area which is satisfactory for a given speed and horsepower rating under favorable conditions. The merit of a design is thus essentially proportional to the "effective area" which, in turn, is determined by the size and number of the restrictions. Abrupt changes in area should be eliminated, if possible, and the area of all restrictions made as large as possible, particularly the "bottle-neck", i.e., the restriction with the smallest area.

Double-End Ventilation Factors

Requisite effective area can be obtained more easily with double-end than with single-end ventilation. Also, auxiliary fans can be avoided to a great extent. In determining double-end ventilation, calculate one end only, and double the result. Designs of the type suggested by Drawings *c,d,e,* and *h* of *Fig.* 6 should be considered as alternatives to single-end ventilation. These designs permit the use of large-diameter, shorter electromagnetic elements for the same overall diameter, hence effective cooling is facilitated. It is difficult to ventilate long motors adequately because the fan diameters, and thus the cubic feet of air per minute, do not increase, although the electrical outputs, and therefore losses, increase tremendously. Designs of the types suggested by Drawings *f* and *i* are required.

Additional precautions should be taken in design calculations for long motors. Fans and air passages should be large enough to give adequate air flow in cubic feet per minute, as expressed by the formula:

$$Air\ flow = \frac{f}{15 \times n} \times d \times A$$

where f is frequency (cps), n is number of poles, d is fan diameter (inches), and A is equivalent area (square inches). This formula generally will indicate the flow within about 10 per cent. Although the air passage should not be too small, the air passage over the shell on a long motor should be small enough to give good air velocity. A velocity over the shell as low as 500 fpm is relatively ineffective; 1000 fpm is good and 2000 fpm is excessive. Less than 15 cfm per horsepower is poor ventilation and more than 25 is probably extravagant. Drawing *j* of *Fig.* 6 shows the best design for single-end ventilation, but Drawing *g* shows a design which is sometimes adequate for low horsepower ratings.

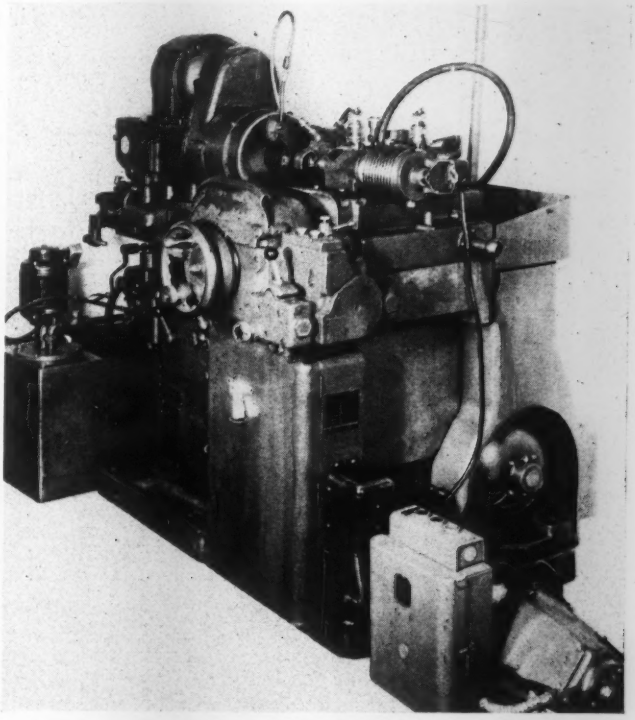
Baffles (air deflectors, shrouds) should be used as shown. The conventional types, of steel, like those illustrated are effective. Distance between the bearing housing and the stator coils usually should be divided by the baffle into two passages approximately equal in area, and the inner end of the baffle should approach closely (⅛-inch to ¼-inch) to the fan blades, roughly midway between the outside and inside diameter of the rotor fans.

At high speeds the ventilation system may be limited by the mechanical strength of the fan or by noise or power considerations. As the speed of a fan increases, the centrifugal force imposes strains on the material that vary with the square of the speed. The power to drive a fan varies with the cube of the speed, and the designer must be careful not to put too much load on the motor. If tip velocity is too high, a fan will produce noise. These various factors tend to limit the diameter of the fan and, therefore, the volumes of air.

Those who are experienced in air-flow calculations may prefer to use their own calculation methods. The empirical formulas in the foregoing are inconsistent with exact theoretical laws of fluid flow, but the errors are not important in practical problems. Effective use of 15 cfm per horsepower generally will give adequate ventilation for motors with normal efficiency used on normal loads.

One broad overall point should not be missed. In the design of a built-in motor, leave room for modi-

Fig. 4—Specially designed shell-type motor built into this grinder makes extremely high spindle speeds possible

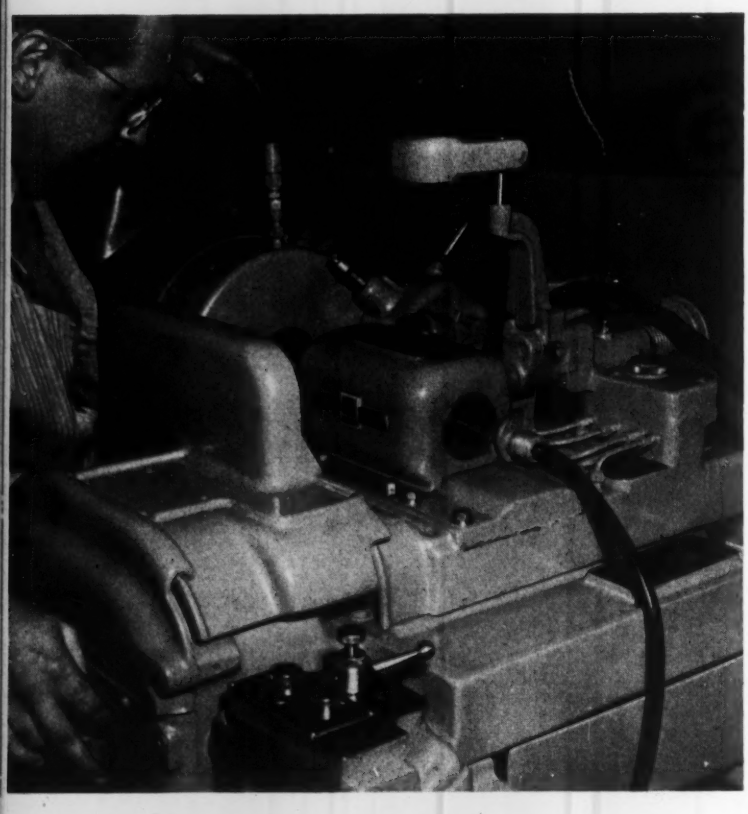


fications and variations. Make sure that there is no chance for increased horsepower requirements, speed requirements, etc., for a design utilizing minimum space. Designers should not cramp themselves. For example, if a design is made for 4 poles, 1800 rpm, will there ever be a chance of going to 2 poles, 3600 rpm, or 2 poles, 1500 rpm, 25 cycles? A two-pole motor cannot be made with the same parts without increasing winding losses considerably. Future trends should be kept in mind when the layout is made.

TOTALLY ENCLOSED, FAN-COOLED MOTORS: The basic engineering facts pertaining to the design of a good enclosed, fan-cooled ventilation system cannot be codified and condensed into simple rules. Electromagnetic parts for open motors are not always satisfactory with an enclosed fan-cooled construction. However, velocities of air over the cooling surfaces should be kept high. Good contacts should be made between the stator shell and the frame to keep internal drops as low as possible. Fans should give enough air to hold the temperature rise of air between 10 and 15C, and of course cooling air should be directed properly to obtain effective results.

Final proof of a ventilation design is a heat run. Thermocouples inserted in contact with the stator coils give reliable results. Thermocouples or thermometers on the frame or on the outside of the tape or compound on the coils give optimistic readings. Core temperature is not an adequate criterion. Possibly it is best to use the resistance method for determining temperatures where tests are not made in a testing laboratory, where:

Fig. 5—High-speed machining heads are driven by high-frequency motors which are powered by inductor-type frequency changers



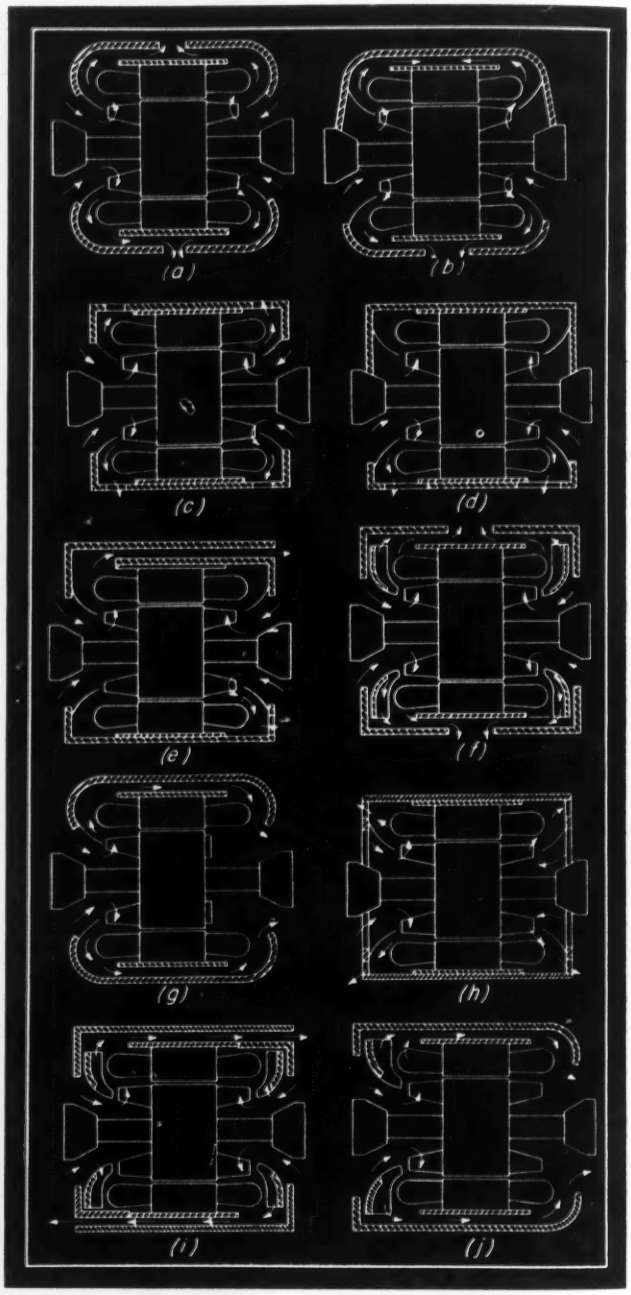
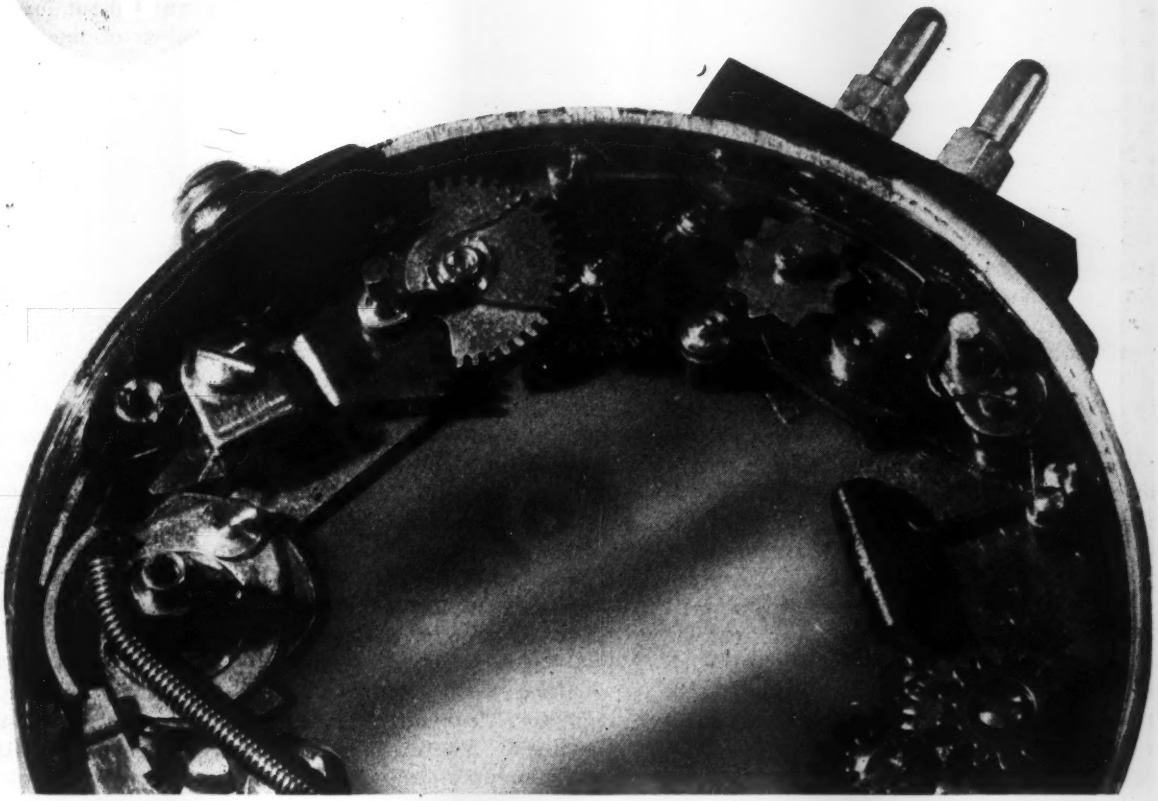


Fig. 6—Ventilation schemes for shell-type induction motors

$$\text{Temperature rise} = (R_2/R_1) \times (234.5 + T_1) - (234.5 + T_2)$$

where R_1 = cold resistance, R_2 = hot resistance, T_1 = temperature of coils when cold resistance is measured, and T_2 = temperature of in-going air at end of test. Resistance should be measured accurately, and extra care should be exercised in determining the

(*Continued on Page* 212)



Designing Instrument Gearing

Basic considerations necessary to observe in the design of successful fine-pitch instrument gearing

By Louis D. Martin
Gear Engineer
Camera Division
Eastman Kodak Co.
Rochester, N. Y.

NEARLY every technical field contains a vast number of mechanisms in which gear trains of various degrees of importance are used. Obviously, the scope of such mechanisms is considerable as is the variety of gearing employed in them. This article can only generalize on the subject and will concern that type of gearing designated by the American Gear

Manufacturers Association as fine-pitch gearing of involute form. Instrument gears can therefore be defined, for the purpose of this article, as involute gears of fine pitch which are used, essentially, to transmit motion rather than power.

TYPE OF DRIVE: In designing a gear train the following factors, in order of their importance, should be considered by the engineer. First, the ratio and kind of drive necessary to obtain the correct speed and direction of motion must be determined. From this a choice of kind of gearing to be used is made.

Spur Gears: Use of spur gears is indicated: (1) When the shafts are parallel, (2) for speeds generally under 1000 surface feet per minute, (3) when there are no high impact loads, (4) when no special provisions are made for thrust, (5) where no fast reciprocating parts are attached to the shafts of, and driven by, the gears such as cams, cranks, pistons, and similar components, (6) where a high degree of quietness or extreme smoothness of operation is not required, and (7) when cost is a factor. Spur gears can generally be produced more cheaply than any other type.

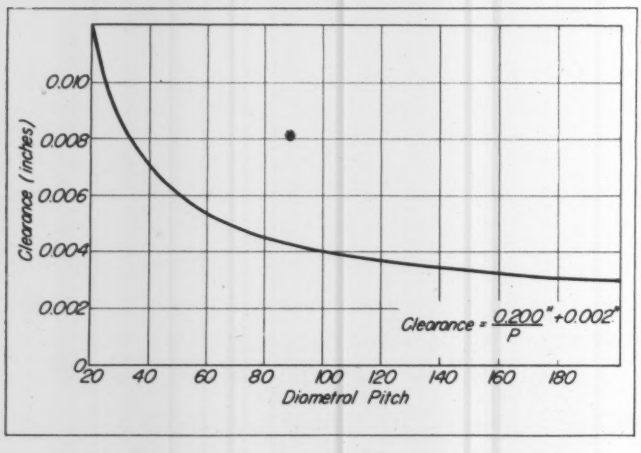
Helical Gears: Use of helical gears with parallel shafts is indicated: (1) Where a high degree of quietness and smoothness of operation is required, (2) generally, for speeds exceeding 1000 surface feet per minute, (3) where fast reciprocating parts must be driven by gears, (4) where adequate thrust provision can be provided, and (5) where there are relatively high impact loads. Helical gears used with crossed axes are not recommended as a substitute for other types of more efficient drives such as bevel gears. When used in this manner, except where the load carrying capacity is nil, they invariably prove troublesome.

Bevel Gears: For right-angle drives where the shafts intersect, bevel gears are indicated. There are several types of bevel gears—straight bevels, spiral bevels and Zerol. Zerol bevel gears are really spiral bevel gears with "zero" spiral angle. In the fine-pitch instrument field spiral bevel and Zerol gears are used to a much greater extent than straight bevel gears. The reasons for this are that they can be produced at lower cost, operate more quietly, and can tolerate some misalignment in their axes without impairing their operation. Adequate thrust provisions must be made when using spiral bevel gears.

Face Gears can be used on right-angle drives where the loads are relatively light. Face gears usually engage a spur pinion which can be moved on its axis and across the face of the gear without loss of efficiency. For this reason the problem of mounting is simplified.

Hypoid Gears: These are used for right-angle drives where the shaft axes cannot intersect. They have all the virtues of spiral bevel gears. In addition, they usually have a greater load carrying capacity.

Fig. 1—Chart of clearance values for fine-pitch gears, 20 to 200 diametral pitch



Face Gear Ratios Restricted

Offset Face Gears: Offset face gears are used for right-angle drives when the shaft cannot intersect. Their load carrying capacity is approximately the same as "on-center" face gears. They may be used for drives where the loads are relatively light. Like the on-center type, the pinion can be moved on its axis without impairing tooth action. In this type of gearing the choice of ratio is somewhat restricted. This applies to both the on-center and off-center types. Ratios of one-to-one are not practical and the pinion must always be smaller than the gear. The most practical applications are those where the pinion is at least one and one-half times smaller than the gear. Face gears also have greater restrictions on face width than bevel gears of equal pitch.

Worm Gears are used when high reductions are rerequired. There are several types of worm gears, the most common being the type in which the gear partially envelops the worm. In another type the worm is shaped like an hourglass and is usually made as an involute helicoid. The mating member is a helical gear. The amount by which the worm envelops its mating helical gear varies with the ratio, pitch and pressure angle. There is still a third type of worm gearing fairly common in coarse-pitch gears and which is finding applications in some of the coarser of the fine-pitch gears. This type has double envelopment, that is, both the gear and worm partially envelop each other, and is known as "Cone Gearing."

Having selected the type of gearing needed for a particular drive, the designer must next make a choice of: Material, maximum fiber stress, pounds per square inch = s; diametral pitch = P, or circular pitch, inches = p; width of face of gear, inches = f; and tooth form or tooth form factor = y. Given in this order, these fundamentals outline the pattern of this familiar Lewis formula $spfy = W$ = transmitted tooth load in pounds, is used for determining the beam strength of gear teeth. While this formula is conservative and is subject to modification by speed and accuracy factors it serves its purpose at this stage of the design process.

TOOTH FORM SYSTEM: Next thing the designer must do is choose the tooth shape best suited for instrument gearing. Fortunately tooth form standards are now available for instrument gears of fine pitch.

AGMA Standard 470.01: This standard establishes clearance for all 20-degree pressure angle, fine-pitch gears of 20 diametral pitch and finer, except bevel gears. Greater clearances than for coarse-pitch gears are required to allow for: (1) Greater wear on point of generating tool, (2) fillet radius produced by dull generating tool which is greater than that derived from formula $0.157/P$, and (3) accumulation of foreign matter at bottom of teeth. The proportions are: Working depth = $2.000/P$; whole depth = $(2.200/P) + 0.002$-in.; clearance = $(0.200/P) + 0.002$-in.; and whole depth factor = $2.200 + (0.002 \times P)$. Maximum recommended tolerance for cutter addendum is plus 0.2 of the clearance, minus nothing. In *Fig.* 1 are charted the clearance values for fine-pitch gears which have been defined by the AGMA as gears of 20 diametral pitch and finer. Whole depth factor applied to fine-pitch gears at specific diametral pitches is given in *Fig.* 2.

AGMA Standard 207.01: This standard covers fine-pitch gears of involute form, shows the basic rack used for this system and gives standard tooth proportions. Table 2 of this standard lists the decrease in gear diameter, the circular thickness (for no backlash), and the minimum number of teeth to avoid undercut in the gear; Data Sheet 1 gives the theoretical center distance at which two enlarged pinions will engage with no backlash; Data Sheet 2 gives the theoretical backlash obtained with two enlarged pinions; Data Sheet 3 gives the formulas for the calculations on which Data Sheets 1 and 2 are based. The formulas can also be used to obtain similar information when an enlarged pinion is meshed with a standard diameter gear. Data Sheet 4 contains a graph and explanations for finding enlargements for helical pinions. Data Sheet 5 gives the tolerance in outside diameter for fine-pitch gears.

Pinion Enlargements: In order to avoid objectionable undercut in pinions with small numbers of teeth, and also to increase the amount of involute action, pinions with 17 teeth and fewer are enlarged as noted in Table 2 mentioned previously. The diameter increments are the same as in the American Standard for 20-degree, full-depth teeth. The circular thicknesses listed in Table 2 are those which are obtained by engaging the enlarged pinion with the theoretical rack at standard depths.

CENTER DISTANCES: When an enlarged pinion is to be meshed with a gear at the center distance which is standard for the numbers of teeth, the gear diameter must be decreased by the amount of the pinion enlargement. When two enlarged pinions or an enlarged pinion and a standard-diameter gear are meshed together, the center distance must be greater than standard.

Standard Center Distance System (long and short addendums): In this system the center distance is made standard for the numbers of teeth, and the pressure angle remains constant. The outside diameter of the gear is decreased the same amount that the pinion diameter is increased. If the number of teeth in the gear is too small, however, undercut will result. The advantages of this system are:

1. No change in center distance or ratio is required
2. Operating pressure angle remains standard
3. Slightly greater contact ratio is obtained than when the center distance is increased.

The disadvantages are:

1. The gears as well as the pinions must be changed from standard dimensions
2. Pinions having fewer than 18 teeth cannot be satisfactorily meshed together
3. In most cases where gear trains include idler gears, the standard system cannot be used.

Enlarged Center Distance System: When two enlarged pinions, or an enlarged pinion and standard-diameter gear, are meshed together the center distance must be increased. For fine-pitch gears, it is usually satisfactory to increase the center distance by an amount equal to one-half of the enlargements. This is an approximation, as theoretically there is a slight increase in backlash. With an increase of center distance, there is a slight increase of operating pressure angle, which is usually not important. The advantages of this system are:

1. Only the pinions need be changed from standard dimensions
2. Pinions having fewer than 18 teeth may engage other pinions in this range.

The disadvantages are:

1. Center distances must be enlarged over standard
2. The operating pressure angle increases slightly with different combinations of gears and pinions
3. The contact ratio is slightly smaller than that obtained with the standard center distance system. This consideration is of minor importance as in the worst case the loss is only about 6 per cent.

How to Specify Gear Centers: Having determined

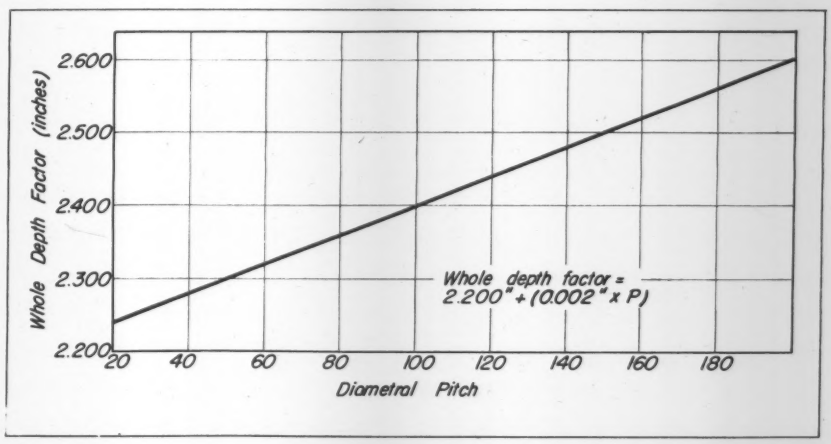


Fig. 2—Right—Whole depth factor for fine-pitch gears at specific DP's

the kind of gear needed for a drive, its ratio, pitch and tooth form, the engineer must next establish the gear centers. The dimension placed on the gear housing drawing must be the nominal center distance plus a working tolerance. This tolerance may be as little as 0.00025-inch or as great as several thousandths, depending on the application. It should always be in a *plus* direction to be on the safe side. The center distance for the gears is one half the sum of their *operating* diameters.

In involute gears there are a number of concepts for pitch diameter. For practical purposes and for the purpose of this article three definitions are given:

1. *Standard Pitch Diameter* is found by dividing the number of teeth by the diametral pitch or N/P. Also by multiplying the circular pitch by the number of teeth and dividing by π, or pN/π
2. *The Operating Pitch Diameter* is the pitch diameter at which the gears operate
3. *The Testing Pitch Diameter* is the diameter at which the gears are tested either with each other or a master. This diameter is usually smaller than the operating diameter to allow for backlash in the gears.

The most satisfactory manner of specifying tolerances for gear centers on drawings is shown in *Figs.* 3 and 4. *Fig.* 3 shows one gear center with the tolerance given in a plus direction. The two legs of the right triangle give the calculated (abbreviated "calc.") dimensions and the angle subtended by the base and the hypotenuse of the triangle with no tolerances. In *Fig.* 4 is shown the way to dimension a number of connected gear centers. The tolerance is put only on the dimension that is to be controlled, namely, the gear centers. The toolmaker, in making the boring fixture, focuses his attention on one thing, the gear centers. All the other dimensions, however, are required whether a jig borer or turntable is provided.

No attempt to give the other related tolerance deviations is made on the other dimensions, since by so doing he would be restricted needlessly. The important thing is the *center distance*. The dimensions specified as "calc." can vary and are put on the drawing for the purposes of general guidance only.

This same philosophy applies also to establishing centers on other types of gearing. In bevel gears, for instance, the distance from the surfaces where the gears are mounted to the intersection point of the shaft axes are dimensioned and the tolerance given in a *plus* direction. Other dimensions that are needed by the machine shop to *hold* the mounting distances should be shown as calculated dimensions. Alignment of shaft axes is very important and tolerances for misalignment in two planes should be given. For spur and spur helical gears the following may serve as a guide:

> For commercial gearing where the gears are not highly stressed and where noise and load requirements are low, misalignment of axes 0.0005-inch per inch of bearing length not to exceed 0.002-inch at any length in either direction of a transverse plane.
>
> For precision gearing, misalignment of axes in either direction of a transverse plane 0.002-inch per inch of length not to exceed 0.0005-inch at any length.

These, of course, are necessarily general tolerances. Many unknowns affect the final decision, such as ratio, face width, material, etc.

FINE-PITCH GEAR TOLERANCES: AGMA Standard

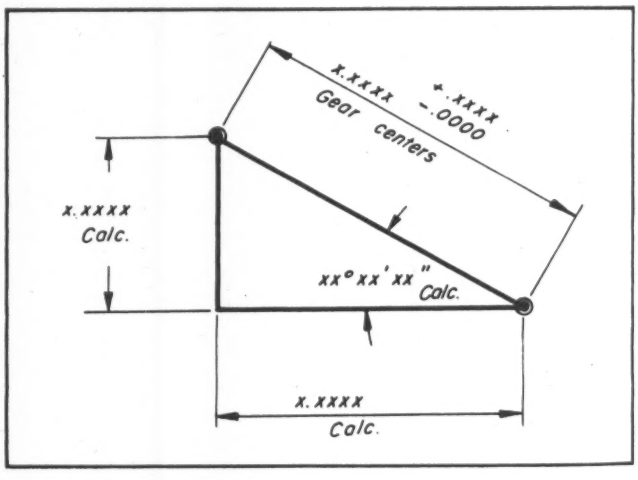


Fig. 3—Above — Proper method for specifying one gear center distance on engineering drawings

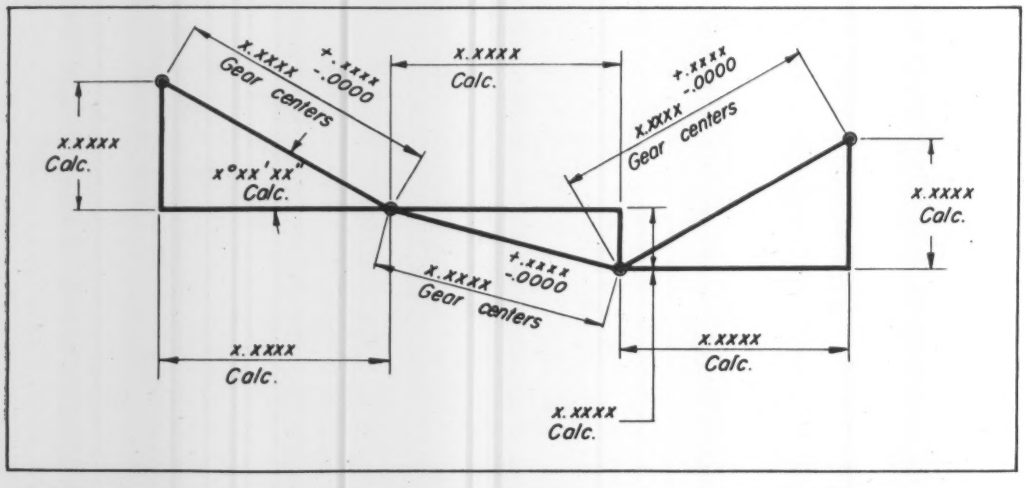


Fig. 4 — Left—Most desirable method for specifying a number of connected gear centers

236.01 covers the inspection and tolerances of fine-pitch gears. This standard was developed to suit the particular needs of fine-pitch gearing which has a multitude of diverse applications. Its aim was to safeguard the interests of quality control in mass production found in this field by developing a clear unmistakable means of specification. Quality specifications are based on the results of the composite check, and do not involve backlash as a factor. Quality is independent of backlash since backlash does not directly affect the accuracy of the gear operation and may be produced or eliminated in the mountings. A choice of backlash to suit the particular application is provided in this standard by giving each class group several ranges of backlash. This provides the flexibility necessary to meet the diverse applications.

Classes: Two class groups were arbitrarily chosen: (1) Commercial, and (2) precision. Each group contains three classes, each based on the maximum permissible error, the allowable backlash value being given by a letter suffix and being applicable to any one of six classes. The classes are tied in with a definite method of inspection which is as follows:

The gears are checked in intimate contact with a master of known accuracy on a suitable variable center-distance fixture, *Fig.* 5. This fixture is provided with a device for measuring or recording the resulting radial displacements or variations during the rotation of the gear. Excluding the effect of backlash this check approaches the action of the gear under operating condition. This type of check is called the *composite* check and gives the combined effect of runout, pitch error, tooth thickness variation, profile error, and lateral runout (sometimes called wobble).

Commercial gears: These are gears that can usually be produced on generally available gear cutting machines and tools in good operating condition without resorting to special techniques or subsequent refining operations. The following tabular values give the tolerances, in inches, for the respective class.

Class	Total Composite Error	Tooth-to-tooth Composite Error
Commercial 1	0.006	0.002
Commercial 2	0.004	0.0015
Commercial 3	0.002	0.001

Precision Gears: These are gears that will usually require the best available precision gear cutting machines and tools in good operating condition. Subsequent refining operations such as shaving, grinding, lapping and sometimes special techniques may be required. Tabular values give the tolerances, in inches, for the respective class.

Class	Total Composite Error	Tooth-to-tooth Composite Error
Precision 1	0.001	0.0004
Precision 2	0.0005	0.0003
Precision 3	0.00025*	0.0002

* This may be the result of selection and segregation.

Backlash Specifications: As already mentioned, backlash in fine-pitch gears is specified by a letter suffix to the quality classification, such as, Commercial 1-B or Precision 3-C. Backlash may be arbitrarily selected to suit the application. Backlash values are

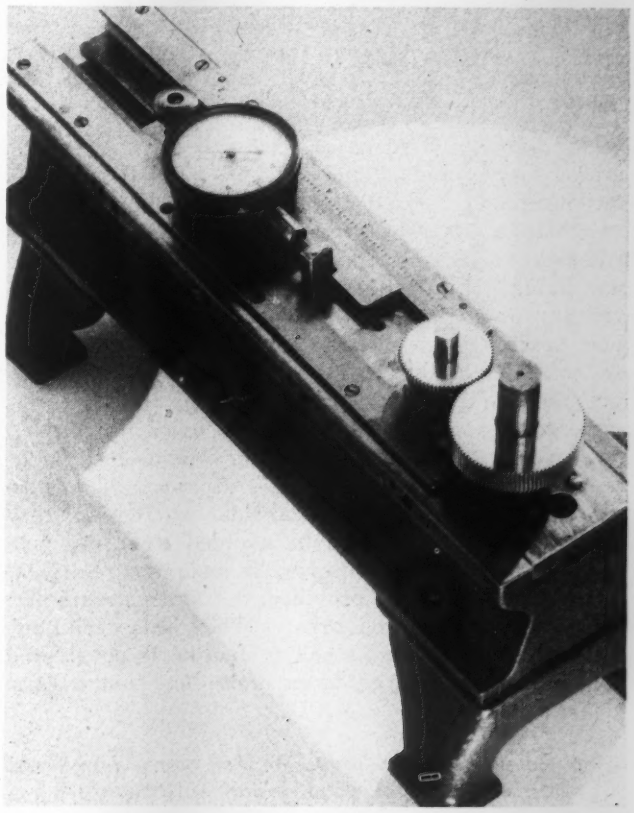


Fig. 5—Typical variable center-distance fixture for checking various types of instrument gears

understood to be between two assembled gears at their *tightest* point of mesh. Thus backlash will be increased when the low points of runout are in contact. Specified backlash, in inches, for different classes of gears are as follows:

Diametral Pitch	Backlash
Class A	
20 to 45	0.004 to 0.006
46 to 70	0.003 to 0.005
71 to 90	0.002 to 0.0035
Class B	
20 to 60	0.002 to 0.004
61 to 120	0.0015 to 0.003
121 and finer	0.001 to 0.002
Class C	
20 to 60	0.001 to 0.002
61 to 120	0.0007 to 0.0015
121 and finer	0.0005 to 0.001
Class D	
No measurable backlash at any pitch	

The relationship of backlash to change in center

distance is proportional to $2 \times tan\phi_t$ and thus for 0.001-inch change in center distance the change in backlash is approximately 0.0007-inch. Following is a table of the equivalent change in center distance, in inches, for a given amount of backlash, in inches:

Change in Center Distance	Backlash
0.006	0.0044
0.004	0.0029
0.002	0.0015
0.001	0.0007
0.0005	0.0004

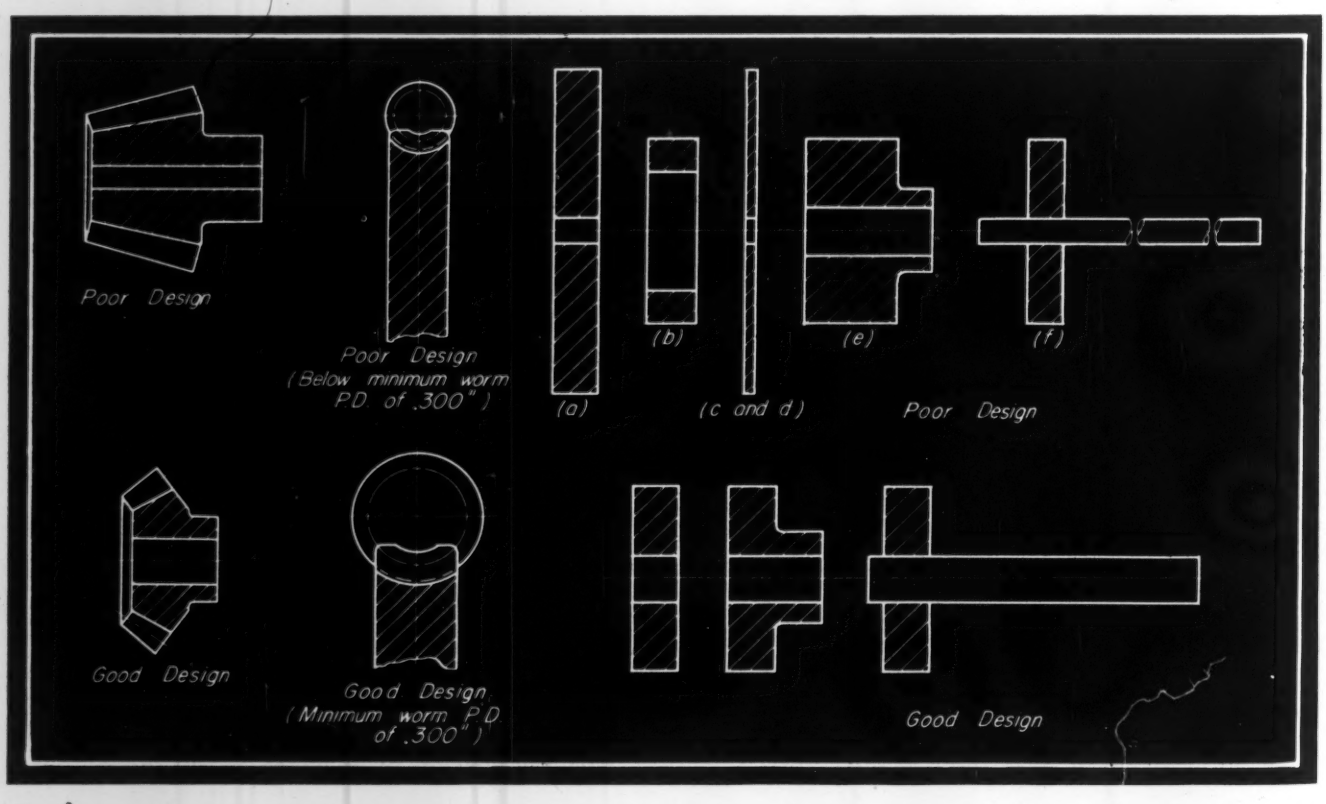
Gear Blanks: The poorest economy in the manufacture of gears, whether fine or coarse pitch, is to start off with poorly designed or incorrectly dimensioned gear blanks. The accuracy to which gears can be produced is largely affected by the design of the gear blank. Unless the gear blank has the required accuracy and rigidity it penalizes the production of good gears. When a gear is designed with a hole, the hole should be large enough to adequately support the blank during the machining of the gear teeth and yet not so large as to cause distortion. Face widths should be wide enough, in proportion to the outside diameters, to avoid springing and to permit obtaining flatness in cardinal surfaces. Very short bore lengths should be avoided wherever possible. It is possible, however, to machine relatively thin blanks in stacks provided the surfaces are flat and parallel with each other. When this technique is employed the whole stack of blanks is considered as a single blank rather than individual components.

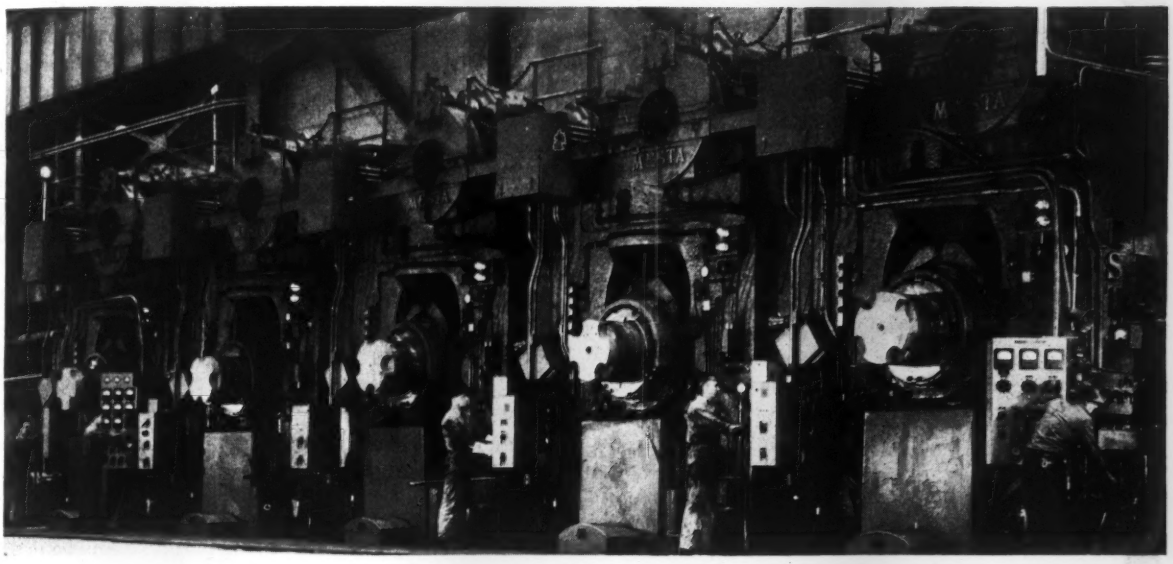
Quite frequently gear blanks must be designed with hubs. Attention should be given to wall sections of the hubs. Too thin a section will not permit proper clamping of the blank during the machining operation and may also affect proper mounting of the gear. Another fairly common practice is that of producing pinions, and on occasion gears, integral with their shafts. This practice lends itself to increased accuracy if the proper precautions are taken. Among these precautions is that of avoiding deflection of the shaft by having the shaft length and shaft diameter well proportioned to the gear or pinion diameter.

In *Fig.* 6 some examples of well designed and poorly designed gear blanks are shown. Most common faults in gear blank designs, *Fig.* 6, which have proved troublesome are: Extremely small bores compared to the outside diameter (*a*), extra large bores compared to the outside diameter (*b*), narrow face widths with large outside diameter (*c*), short bore length (*d*), thin hub walls (*e*), and gears integral with shafts that lack stiffness (*f*). It is well recognized that the cost aspects often tempt the designer to select the type of gear blank he knows is not ideal. On occasion, compromises must be effected. It is well to remember that in the final analysis a poorly designed gear blank is not the most economical one.

The complete story of the design of fine-pitch instrument gearing can hardly be told in the space allotted. Naturally, there are many gaping holes in this tale but it is hoped that attention of the designers has been focused in as clear and logical a manner possible on a few of the major points that must be considered in successfully designing fine-pitch instrument gearing.

Fig. 6—Below—Common faults in gear blank design and examples of good design practice





Variable-Speed Drives

Recent developments involve increase in speed range and more precise regulation

By E. L. Schwarz-Kast
Armour Research Foundation
Chicago

DURING the last few years many improvements in electrical variable-speed drives have made possible remarkable refinements in machine drives. A brief survey, therefore, of some of the more important developments, especially those which have been designed since publication of the article, "Selecting Drives for Speed Control" in the June, 1945 issue, will be

presented. Indicative of the precise control possible Regulex exciters provide instant and automatic regulation of tension, speed and strip gage in the steel mill shown in the head illustration.

SHUNT MOTOR WITH FIELD CONTROL: The practical range of speed increase by field control in the past was a maximum of 1:4. This figure has now been revised. Westinghouse Electric Corp. developed a shunt motor with an adjustable speed range, through field control, of 1:8, and with excellent regulation at all speeds. The main factors which limit the speed increase by field control with conventional motors are:

Changes in residual magnetism
Cross-magnetization by armature flux
Demagnetization of the main field by armature flux
Increase of armature-reactance voltage with increased speed.

These factors cause instability or poor commutation when the field is weakened beyond the 1:4 speed range and the operation of a conventional field-controlled motor at higher speeds becomes unsatisfactory. The main reason for the first three of these shortcomings is the low field intensity in the main poles at high speed. The idea on which the development of the new motor is based on obtaining field weakening by only a slight decrease of flux density.

The new Westinghouse motor for 1:8 speed increase by field control is a four-pole machine and differs only slightly from a conventional machine. It has two separate field circuits and a two-circuit armature winding (two brushes), as shown in *Fig.* 1. The upper poles M_1 and M_2 are the main field, and are only slightly weakened by speed control; even at the highest speed, there is still an appreciable strength (60 per cent of full field). V_1 and V_2 are called the variable field which can be varied from plus 100 per cent to minus 38 per cent.

The highest speed is naturally limited by commutation and by the mechanical conditions. This speed limit was found to be 3600 rpm, but stable operation, excellent regulation and satisfactory commutation were obtained over the 1:8 speed range, 450-3600 rpm. This wide speed range motor can substitute successfully for the conventional variable-voltage control with motor generator for a speed range up to 1:8 where d-c power is available.

VARIABLE-VOLTAGE CONTROL: Important developments have taken place in the field of variable-voltage control which is still the most convenient electrical method for very wide speed range. For a better understanding of the new developments it might be appropriate to recall the limiting factors of the conventional variable-voltage systems, as they were discussed in the previous article:

1. The voltage drop in the armature loop causes poor regulation, especially at low speed
2. The obtainable lowest speed is limited by the residual magnetism of the generator. Most commercial generators have a residual voltage of three to four per cent rated voltage.

For compensation of the voltage drop in the loop, various means are available: One involves the addition of a series field to the generator, making a flat compounded machine. As the armature current increases, the series field tends to maintain constant generated voltage.

Another method is the use of a series-wound exciter, with the series field of the exciter inserted in the armature-loop circuit. With increasing load the loop current increases and, in turn, the exciter voltage and the field current of the generator which boosts the generator voltage, improving regulation.

The trend of these methods is toward maintaining a constant generator voltage which improves the reg-

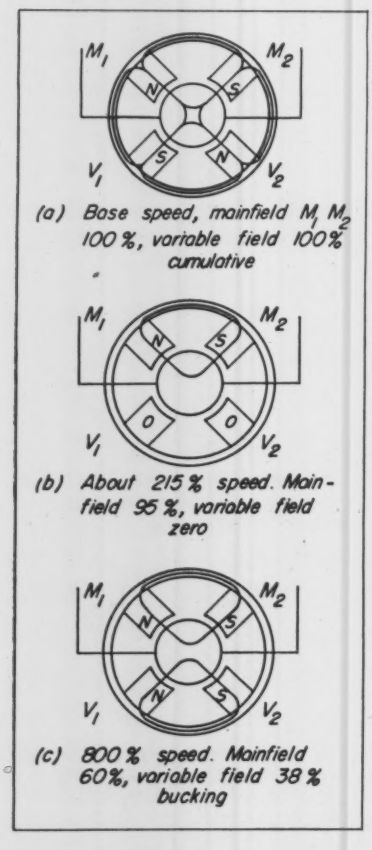
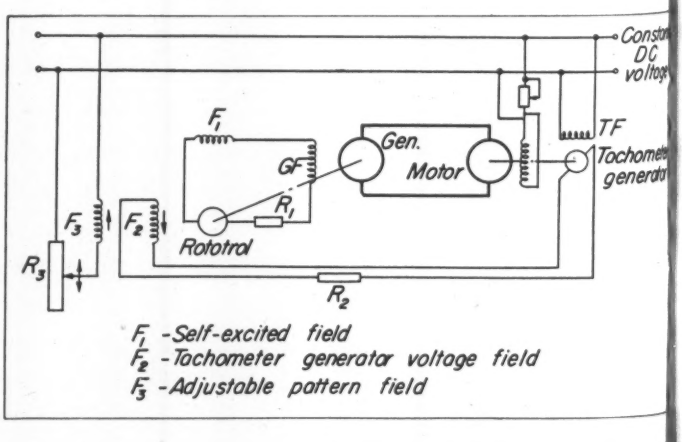


Fig. 1—Shunt motor designed for 1:8 speed range by field control. At A is shown base speed operation with main field 100 per cent, variable field 100 per cent cumulative. Diagram B shows circuit arrangement for about 215 per cent speed, main field 95 per cent, variable field zero. Diagram C is for 800 per cent speed, main field 60 per cent, variable field 38 per cent buckling

Fig. 2—Variable-voltage drive with Rototrol regulating exciter and tachometer generator



ulation of the motor speed. However, even if perfect constancy of generator voltage is established, there still remains the fact that the motor slip varies with the load. This cannot be tolerated in cases where a high degree of speed stability is required. Also, the question of residual magnetism in the generator field remained unsolved.

CLOSED-CYCLE CONTROL: To meet severe requirements in regard to wide speed range, and to accurate and constant speed adjustments, closed-cycle controls have been developed. In such a control a quantity representing the output, in this case the actual motor speed, is measured and balanced against an adjusted pattern, representing the desired speed.

Variable-voltage systems are especially suitable for a closed-cycle control, due to their inherent high amplification factor. The control is effected in the field circuit of the exciter and the power necessary is roughly one per cent of the exciter output which is, in turn, one per cent of the generator output. In other words, the required signal power is about 1/100

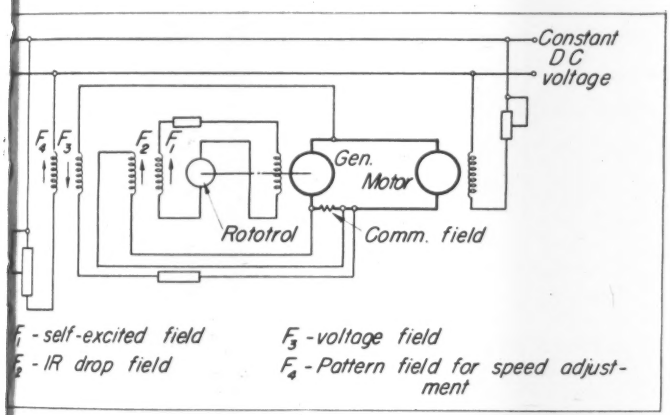


Fig. 3—Variable-voltage drive with Rototrol regulating exciter, IR drip and generator voltage signal

per cent of the generator power, i.e., the amplification factor is 10,000.

The signal for the true motor speed can be obtained in two ways: First, by a tachometer generator, with its field excited by a constant d-c voltage or by permanent magnets. The tachometer generator, coupled with the main motor, develops a voltage which is directly proportional to the motor speed.

Another method of speed indication, without using a tachometer generator, is based on the commonly known fact, that the motor speed is proportional to the counter-voltage (cemf) of the motor. The cemf equals loop voltage minus *IR* drop in the motor armature. Consequently, a sufficiently accurate signal of the motor speed can also be obtained by measuring the loop voltage and *IR* drop.

Aside from good regulation, another important feature of any speed control device is the time of response. In the described closed-cycle control the total time of response is determined by the time lag in the response of the exciter and the generator and by the inertia of the drive. To keep this time of response at a minimum, the time constant of the exciter has to be as small as possible. On the other hand, too short a time of response creates the danger of hunting. A closed system can be "antihunted" by an additional antihunting field on the exciter. This field can be, for example, in series with the secondary of a transformer whose primary is connected across the loop voltage. A sudden change in voltage induces voltage in the secondary of the transformer and the polarity of the antihunting field is so chosen that it opposes the control field. On the other hand, at steady state no voltage is developed in the primary or secondary of the transformer and no current flows through the antihunting field and the system is only controlled by the control field.

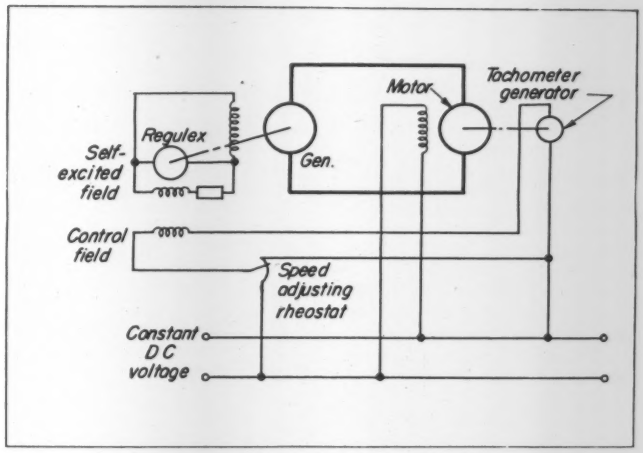


Fig. 4—Variable-voltage drive with Regulex regulating exciter and tachometer generator

REGULATING EXCITERS: The closed-control system, discussed previously, requires the use of regulating exciters. They differ from conventional exciters in the following respects: The field structure is laminated for fast response to changes in field excitation, and to obtain linear response of output voltage to changes in field current they operate only on the straight portion of the magnetizing curve.

When used in a closed-cycle control the regulating exciters are equipped with a multiplicity of field windings, each responding to a particular control signal. The total sum of the fluxes set up by the various fields determines the output voltage of the exciter and in turn the generator voltage and motor speed.

Regulating exciters are built by various companies, such as: Allis-Chalmers Mfg. Co. (Regulex), Westinghouse Electric Corp. (Rototrol), and General Electric Co. (Amplidyne).

The following descriptions and wiring diagrams of some significant variable-voltage arrangements illustrate the practical use of these regulating exciters in closed-cycle controls. As an example, *Fig.* 2 shows the principal connections of a Rototrol as a regulating exciter for a variable-voltage drive using a tachometer generator for speed indication.

The generator field is excited by a Rototrol. The Rototrol is self-excited by the series field F_1 and the

circuit is tuned by adjusting the resistor R_1. Field F_2 is excited by the output of the tachometer generator and has also a resistor R_2 in the circuit for adjustment. Field F_3 is the pattern field which is excited from the constant d-c voltage. The motor speed can be varied by adjustment of the resistor R_3. Field F_2 bucks field F_3. At any adjustment of resistor R_3 a field strength F_3 is established. Field F_3 together with the base field F tends to adjust the motor speed adequately. The tachometer generator will circulate sufficient current in the field F_2 to neutralize the pattern field. When the true adjusted speed is reached the control field F_1 and the tachometer field F_2 cancel each other and the exciter voltage is sustained merely by the self-excited field F_1.

In *Fig.* 3 is shown a variable-voltage drive with a Rototrol, but using the motor counter emf instead of a tachometer for speed indication. The Rototrol is self excited by field F_1 and through the adjusting resistor. If a constant motor speed should be obtained, the counter emf of the motor has to be kept constant, consequently the voltage of the motor has to be increased, as the load current increases, by an amount equal to the IR drop which is measured by field F_2 connected across the commutation pole winding of the generator. Generator voltage regulation is measured by voltage field F_3. The speed is adjusted by the variable rheostat and field F_4. When excitation is applied to the pattern field the exciter voltage builds up until the F_2 and F_3 cancel the pattern field.

When this point is reached, the self-excited field sustains the exciter voltage and the drive operates at a speed corresponding to the adjusted excitation of the pattern field. If the speed changes there is a net difference of excitation to raise or lower the exciter voltage until a balance is reached when the fields F_2 and F_3 again cancel the adjusted-pattern field and the exciter voltage is sustained merely by the self-excited field F_1.

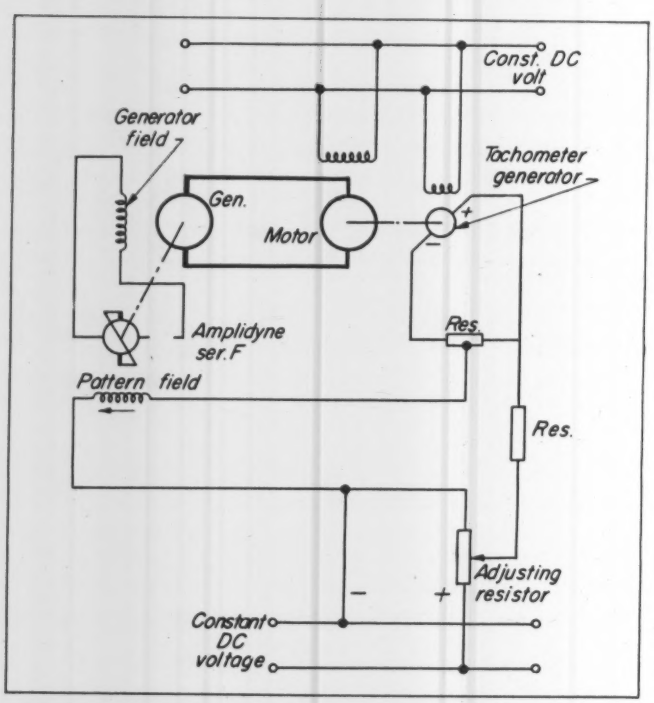


Fig. 5 — Above — Variable-voltage drive with Amplidyne and tachometer generator

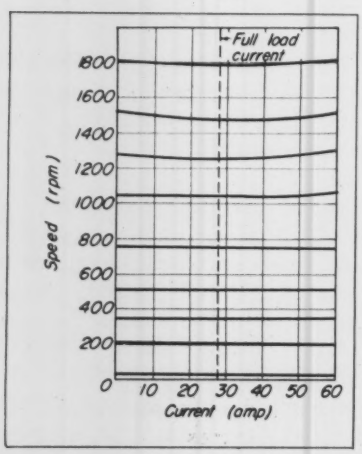


Fig. 6—Speed-load characteristics obtained from tests on a 7½ hp drive using a Rototrol and counter emf speed indication, using the circuit shown in Fig. 3. Speed range 1800 to 15 rpm, 120:1

Has High Amplification Factor

A variable-speed drive with a Regulex regulating exciter is shown in *Fig.* 4. The generator voltage is varied to hold constant motor speed by balancing the tachometer voltage against a d-c reference through an adjusting rheostat. A special form of regulating exciter is the Amplidyne of the General Electric Co. It is a d-c generator which, due to its special electrical design, has a very high amplification factor. A conventional exciter for 10-kw output, for example, requires approximately 100 watts for excitation. An Amplidyne of the same output requires only one watt. This extremely small amount of energy required for the shunt-field excitation permits the shunt-field flux to build up rapidly. As the resistance and reactance of the short circuited armature is very small, armature flux also builds up rapidly. The time required by an Amplidyne to build up its rated voltage from zero is only 1/20 to 1/15-second. *Fig.* 5 shows the diagram of a variable-speed drive using a tachometer generator and an Amplidyne as regulating exciter.

The output voltage of the tachometer generator is applied to a resistor and a portion of the voltage drop across the resistor is used as speed signal. The reference voltage is obtained from a constant d-c source. A portion of this d-c voltage is used as the pattern against which the speed signal is balanced. The current through the Amplidyne pattern field is proportional to the net difference between the reference voltage and the tachometer voltage. The speed of the motor is set by the adjusting resistor.

To obtain increased accuracy of speed control, the above mentioned regulating exciters can be used for any variable-voltage drive instead of a conventional exciter. Wide speed range and closely held constant speeds obtained with a Rototrol are shown in *Fig.* 6. In general, it can be said that by using these regulating exciters, a speed of response, an accuracy of control, regulation and magnitude of speed range can be achieved, which could never be obtained with conventional exciters.

MACHINE *Editorial* DESIGN

Psychology in Design

Does the machine designer give sufficient attention to the whims, physiology and psychological reactions of the average potential operator in developing his machine? Recent studies prove that on the whole the answer is "No." Much advancement evidently is still to be made in the direction of "fitting the machine to the man."

In view of the current urgent necessity for increase in production and improvement in employer-employee relations, too much emphasis cannot be placed on the need for added consideration of the human element in design. It is not enough to develop a machine which, from the standpoint of its performance alone, is well-nigh perfect. Such a machine can fail to reach the anticipated level of production in the field unless adequate provision has been made for the co-ordination of the efforts of the operator with the efforts of the machine.

As Theodore F. Hatch, director of research of the Industrial Hygiene Foundation puts it, "A dual responsibility exists in the design of industrial machines—not only must the machine possess the desired functional characteristics but it must also be correct, physiologically, if the combination of man and machine is to operate together with the greatest success."

Many alert designers have been aware of this dual responsibility and have done everything in their power to adopt methods of minimizing operator fatigue and creating the right psychological reactions on the part of the operator. Much consideration has been given, for instance, to locating hand controls within easy reach. Color has been used to give the machine an attractive appearance and to differentiate between the various controls where these are grouped together. Interlocking controls, as discussed elsewhere in this issue, have been applied to complicated machines not only to take care of sequence in operation but also to protect the machine and insure complete safety for the operator.

Renewed attention is being focused on this aspect of design through the investigations now being carried out by several research institutions. The results should be invaluable to the design field in general and should serve to provide additional scientific data for those designers who already have incorporated "human engineering" in their development work.

L. E. Jerry

Which Wrought Stainless Steel?

By Stanley P. Watkins
Manager

and

Roland J. Berkol
Engineer

Development Engineering Dept.
Rustless Division
The American Rolling Mill Co.
Baltimore, Md.

Part II—Fitting the Material to the Part

AS SHOWN in Part I, the mechanical properties and processing characteristics vary for the several grades of wrought stainless steels and, consequently, different types are best suited for different machine parts. For the sake of consistency, the following discussion of general characteristics and applications of the various grades will cover the types grouped as in TABLES I, II and III.

CHROMIUM-NICKEL TYPES: These have the highest corrosion resistance of all the stainless steels, as well as unusually fine mechanical properties. They cannot be hardened by heat-treatment, yet great tensile strength and hardness may be developed by cold working. Normally nonmagnetic, they become slightly magnetic as they are cold worked. They are widely used, with or without additional alloying elements, in sheet, strip and plate form as well as in bar and wire products.

Type 301: The standard grade for structural purposes, combining maximum strength with maximum workability. Of all the stainless types, "17-7" requires the least cold working or cold reduction for a given strength increase. While slightly less corrosion resistant to certain media than Type 302, this grade is satisfactory for such products as cream separator disks, milk pump shafts, sanitary plug valves, and wire springs.

Type 302: Most widely used of all the grades, "18-8" most closely approaches being an all-purpose stainless steel. When cold worked to high tensile

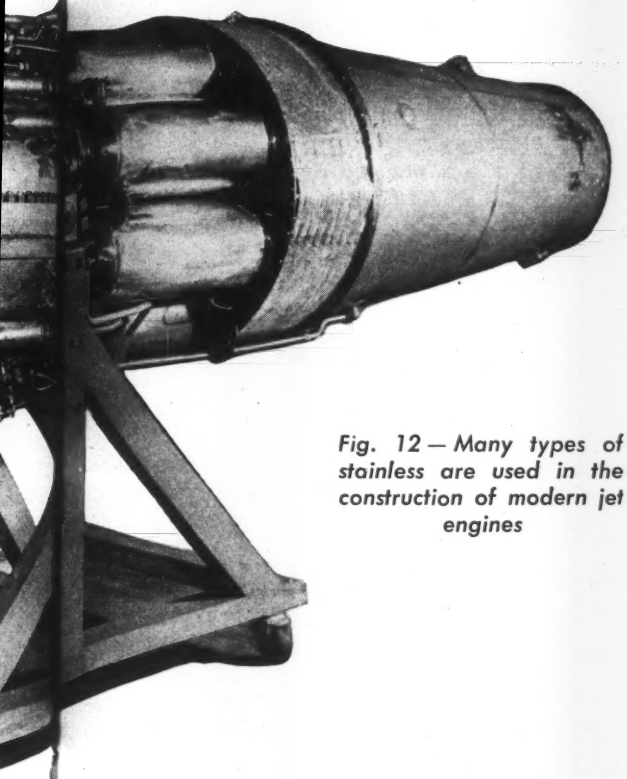


Fig. 12 — Many types of stainless are used in the construction of modern jet engines

strengths, it has a lower ductility than Type 301; consequently, it generally is used in the soft annealed condition. Because of its toughness and great ductility, it can be drawn, spun, and otherwise worked severely in the cold state. It can be readily welded by oxy-acetylene, electric arc, atomic hydrogen, inert-gas shielded arc and resistance welding methods. Welds are generally as tough and ductile as the parent metal, especially in the case of spot welds. It may be machined at speeds about 45 per cent of those for Bessemer screw stock and has good forging characteristics. Type 302 consistently withstands the corrosive effects of many solutions encountered in the textile, paper, chemical, dairy and food handling industries. It resists nitric acid well, halogen acids poorly, and the sulphur acids moderately. For the latter, Types 316 and 317 are suggested. Typical Type 302 applications are machine screws, wire-cloth filters, canning machine parts, valve stems, plugs and seats, mixing shafts, and various stamped, drawn and spun products.

Type 302*B:* Resistance to scaling at elevated temperatures superior to that of Type 302 is imparted to this alloy by the addition of silicon. It is used exclusively for parts which are exposed to high temperature in service. Slightly magnetic when cold-worked, it is nonmagnetic in the annealed condition. Examples of products made from this grade are air preheaters, annealing boxes, furnace parts, range oil-burner shells, still liners, piston rods, shafts, valves, and acid pumps.

Type 303: Several varieties of this grade are offered having free machining properties. Cutting speeds of 75 per cent or more of those used for Bessemer screw stock are attainable. Mechanical and physical properties are similar to those of Type 302. Type 303 is readily ground or polished. It also is readily forged but will withstand only moderate cold work. For deep-drawn or cold-headed parts, Type 302 is preferred. Type 303 can be welded and brazed, albeit with some difficulty, and can also be soldered. This grade has nongalling properties that facilitate machine part disassembly and minimize damage to bearing surfaces. Some of the many end uses of Type 303 are machined wood and cap screws, carburetor bushings and needle valves, pump shafts and parts, valve seats and plugs, etc.

Type 304: This type is similar to 302 except that

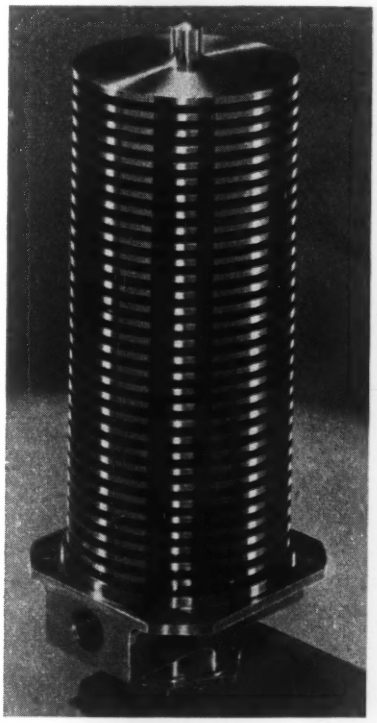


Fig. 13—Right—Flash boiler of airplane heat exchanger for cabin heating made of Type 309 (25-12) stainless to insure high scaling resistance

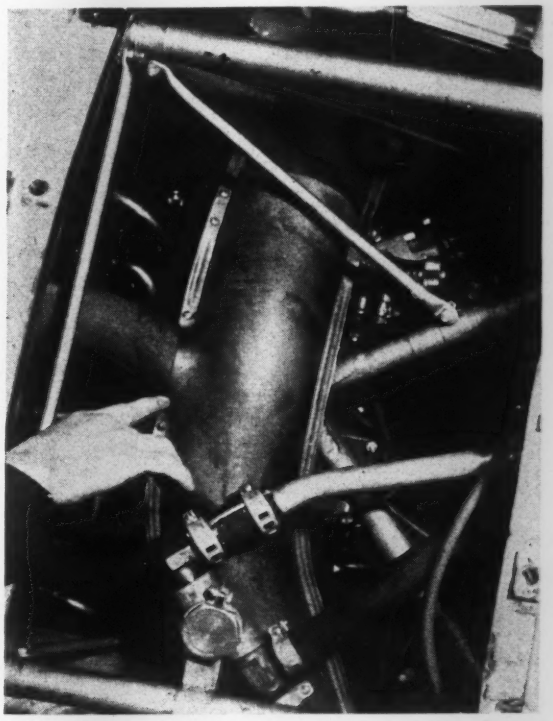


Fig. 14 — Extreme Right— Because this airplane-engine exhaust collector ring must have high corrosion and scaling resistance plus high strength at elevated temperatures, Type 347 (18-10 Cb) stainless steel was selected

it is less susceptible to "intergranular corrosion". It can, therefore, be exposed longer in the critical temperature range of 800 to 1650 F before the chromium content at the grain boundaries is reduced to the danger point. Type 304 has better welding characterisitcs than Type 302. Also, its somewhat higher chromium content provides corrosion resistance slightly superior to that of Type 302. In addition to welded structures, typical end uses for Type 304 are cold-headed, roll-threaded machine screws, thermocouple wells, sanitary fittings, etc.

Type 305: This is an 18-8 type especially developed for spinning. The higher nickel content reduces the work-hardening rate and increases the amount of spinning that can be done between anneals.

Type 308: Analysis and properties of this grade fall between Types 302 and 309. It is used for applications requiring oxidation resistance and strength at high temperatures superior to those which can be handled by Type 302. Type 308 performs well in drawing and forming operations. Welds are easily made and are strong. A considerable tonnage of this grade is supplied in the form of core wire to welding electrode manufacturers. It is used both for Types 308 and 347 electrodes.

Types 309 *and* 310: These are primarily heat-resistant grades, known as "25-12" and "25-20", respectively. They are ductile, machinable, and produce strong welds. Both are used in applications requiring the combination of superior scaling resistance and exceptional high-temperature tensile and creep strengths. Type 310 offers greater resistance to scaling than Type 309 due to its higher chromium and nickel contents. Both are used for welding electrodes. Type 309 is used for furnace bolts and aircraft heating boilers, *Fig.* 13, while Type 310 is used for diesel injector nozzles.

Type 314 is for applications requiring the corrosion resistance of Type 310 where even more severe scaling conditions are encountered. Typical end uses for this grade are furnace doors, skids, heat exchangers, pump parts, piston rods, and valve parts.

Types 316 *and* 317: These grades were designed to improve the corrosion resistance of the 18-8 types by the addition of molybdenum. Also, nickel content is increased to make the alloys more workable. Their advantages are: (1) Improved corrosion resistance, especially for use in the textile, paper and chemical industries. Phosphoric, sulfuric and acetic acid solutions have a strong tendency to pit the conventional 18-8 stainless steels, whereas these types resist this tendency; (2) the highest creep strength

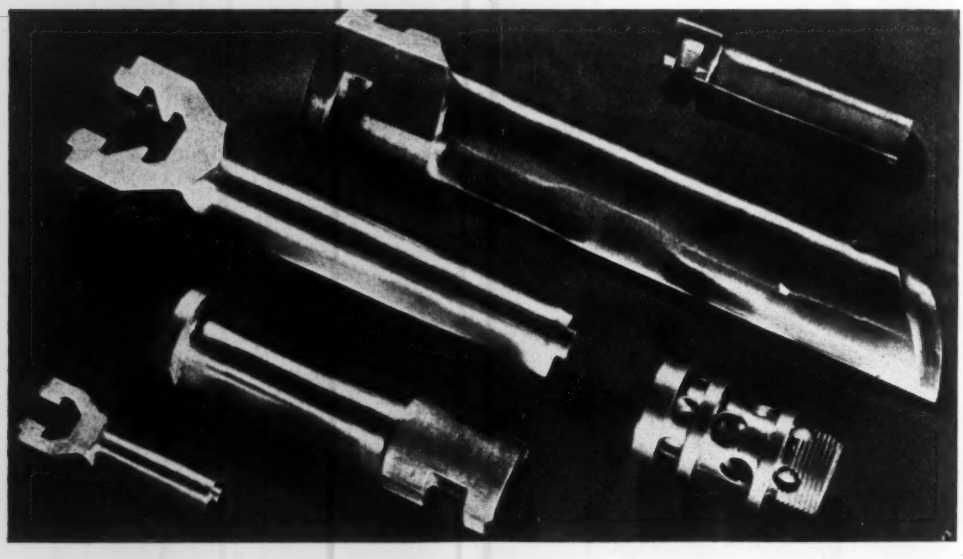
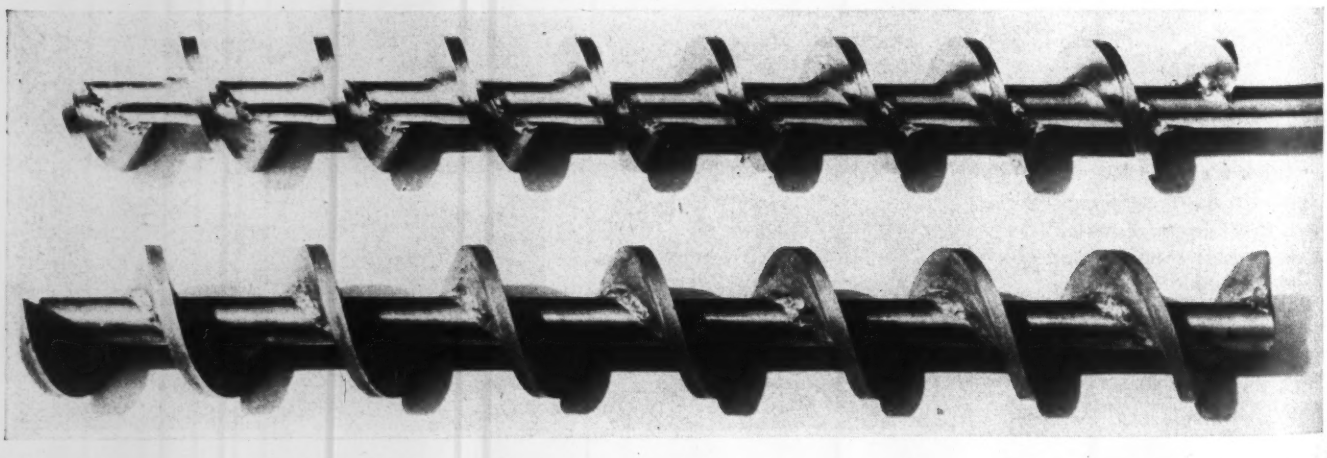


Fig. 15 — Left — These steam turbine blades and similar parts were machined from Type 403 stainless because it offers good strength at 850 F plus resistance to condensing steam and erosion

Fig. 16—Below—Domestic coal stoker feedscrews of chromium stainless (generally Types 410 or 430) have great strength; combine resistance to corrosion and abrasion



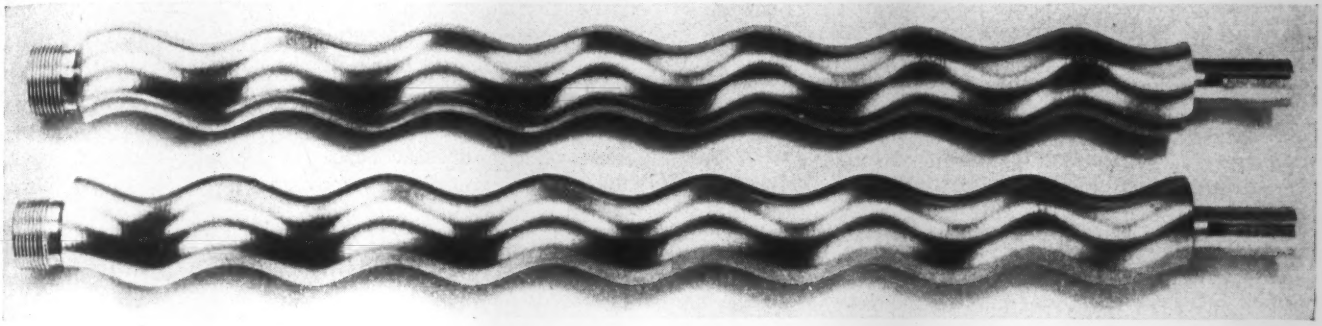
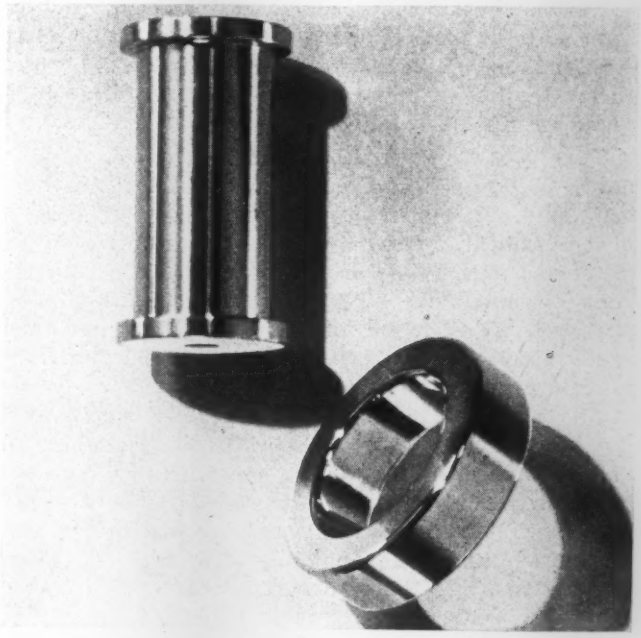


Fig. 17—Above—Type 416 stainless was chosen for these water-pump rotors because of its low cost, excellent machinability and good corrosion resistance

Fig. 18—Below—Counterweight pin and bushing for aircraft engine are of Type 440C stainless, hardened for wear and corrosion resistance



at elevated temperatures of the standard stainless grades; and (3) high short-time tensile strength at elevated temperatures. Some of the end uses are centrifuge screens, pickling chains, paper-tube mandrels, film developing machine parts, paper pulp digestor fittings, and shafts for acid pumps.

Types 321 *and* 347: These are known as the "stabilized" grades and are not subject to intergranular corrosion encountered with the other members of the 18-8 family. These types are used optionally where service conditions are too severe to permit use of welded low-carbon 18-8 (type 304).

Types 321 and 347 are used extensively for exhaust collector rings (*Fig.* 14) and exhaust stacks on airplane engines. Due to their "stabilized" nature, these grades retain their resistance to the corrosive attack of ethyl gasoline exhaust gases when the engine is running, and to the mixture of the products of combustion and the condensate that forms after the engine cools. Both types lend themselves well to deep drawing and drop hammer operations. In addition to the above, typical applications are aircraft engine exhaust flanges (both grades), chemical equipment and furnace bolts (Type 321), jet engine and gas turbine parts, nitric acid pumps and valves, resin kettle mixer arms, and welding electrodes (Type 347).

MARTENSITIC CHROMIUM TYPES (*hardenable by heat treatment*): All of the types in this group are produced in bars and wire only, with the exception of Type 410 which is also produced in sheet and plate form, some Types 414, 420 and 440 in strip, and Type 431 in sheet and strip. These grades are hardened by heating above the critical or transformation temperature and then rapidly cooling in oil or air. By proper selection of grades and analyses, a wide range of "as quenched" hardness is available, depending primarily on carbon and chromium content.

The term "hardenability" as it applies in the case of low-alloy steels to hardening, has relatively little significance with stainless steels. This is because all hardenable stainless steels harden uniformly throughout, even in large sections, when rapidly cooled from above the critical temperature. The degree to which the stainless steels harden is referred to by the terms "as quenched", "full" or "maximum" hardness.

These grades attain maximum corrosion resistance in the hardened and stress-relieved condition, with most of the carbon dispersed as carbides in the martensitic structure. Only the lower carbon grades should ever be placed in service annealed. All martensitic types are consistently magnetic.

Type 403: This is modified Type 410 especially suitable for steam turbine blades, *Fig.* 15, and other highly stressed parts.

Type 410: The lowest in cost of the truly stainless steels, "12 Chrome" is suitable for use where severe corrosion is not a problem. Type 410 has good mechanical properties, is readily drawn, forged, cold worked and can be machined at speeds approximately 55 per cent of those used for Bessemer screw stock. A wide range of mechanical characteristics may be obtained by heat-treatment, and the best corrosion resistance plus best mechanical properties are obtained in the hardened and stress-relieved condition.

While inferior to Type 430 in resistance to various chemicals, it has certain qualities that give it preference in numerous applications such as in oil refining equipment. Type 410 may be used where dampness, alone or coupled with chemical pollution, causes carbon steel to fail quickly; or where abrasive wear or inaccessibility makes paint protection diffi-

cult, as in grain elevator chutes, coke and coal-handling equipment, and coal sizing screens.

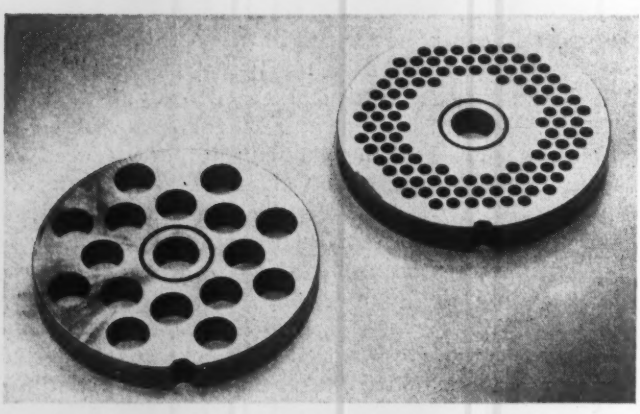
Bar and wire are used extensively in the manufacture of screws and bolts; stems, seats and other parts of valves; pump shafts and rods; pistons; textile bobbin rings; stoker feedscrews (*Fig.* 16); motor shafts; conveyor pins; automobile throttle control wire; forgings and similar applications requiring high strength and good corrosion resistance.

Type 414: Addition of approximately 2 per cent nickel to the Type 410 analysis produces a steel of greatly increased strength in both the annealed and cold worked conditions. This grade, known as "12-2", cannot be softened by annealing to less than approximately 250 Brinell and, consequently, is more difficult to fabricate than Type 410. However, it has slightly higher mechanical properties when heat treated than Type 410 and may be blanked and moderately formed when hardened and tempered. Its corrosion resistance is about the same as that of Type 410. It will resist the corrosive action of many foodstuffs as well as atmospheric corrosion. Some of the better known applications of this grade are tempered rules, scraper knives, and springs.

Type 416: This grade is the free machining "partner" of Type 410 and has similar physical, mechanical, and heat-resisting properties, and response to heat-treatment, with only slightly less corrosion resistance. It is widely used for high-speed production runs on automatic screw machines. Cutting rates approach 85 per cent of those used for Bessemer screw stock. Short, brittle chips are characteristic of this grade of stainless steel. The general workability of Type 416 is fair. It is readily drop forged, but hot trimming is preferable to cold trimming. It can be welded, solders with some difficulty, does not braze well, and is not recommended for severe cold upsetting. Among its many uses are machine screws, bolts and nuts, water-pump rotors, *Fig.* 17, steam valve stems and trim, carburetor parts, dies, gages, magnetic parts, oil-burner nozzles, axles, gears and pinions, and shafts for water pumps, radar and automobile manifold heat-control valves.

Types 420 *and* 420*F:* These high-carbon stainless steels can be brought to high hardness by proper heat treatment. Types 420 and 420F are used for cutlery, dental and surgical instruments, bearings and similar applications requiring high abrasion and corrosion resistance.

Fig. 19—Type 440C stainless, a high-carbon grade, was selected for these meat chopper plates because it has good abrasion and corrosion resistance



Type 420 can be machined at approximately 45 per cent of the speeds used for Bessemer screw stock. The free machining counterpart, Type 420F, has a cutting rate approximately 10 per cent higher.

Not Usable in Annealed State

These grades should *never* be placed in service in the annealed condition, and a chemically clean surface is essential after final heat-treatment. The high-carbon grades never should be pickled in the fully hardened condition. Stress relieving after hardening and before pickling is essential to avoid pickling cracks. These grades can be forged, moderately cold-worked, welded and soldered. In addition to their use for cutlery, dental and surgical instruments, industrial applications include drive and self-tapping screws, spanner wrench inserts, plastic molding dies, valve seats and plugs, springs, shafts, gears, cams, pivots, ball bearings, gages, etc.

Type 431: Known as "16-2," this grade has high strength in both the annealed and cold-worked conditions, as well as the best corrosion resistance of all the martensitic grades of stainless steel. It may be heat treated to about 400 Brinell retaining good ductility, but cannot be annealed to lower than about 250 Brinell. It has been found increasingly useful for aircraft and marine parts, such as forgings, nuts and bolts; also similar applications requiring high strength, toughness and resistance to salt-spray attack.

In addition to the above, Type 431 has been used for precision springs; stems and gates of petroleum valves; outboard motor propeller shafts, surgical trusses, and Jordan bars (in paper production).

Types 440*A*, 440*B*, 440*C and* 440*F:* These high-carbon, 17 per cent chromium grades provide a greater degree of "as quenched" hardness and corrosion resistance than do Types 420 and 420F. Type 440A can be hardened to a range of 500 to 560 Brinell; Type 440B from 520 to 590 Brinell; and Types 440C and 440F from 540 to 620 Brinell. Type 440F, the free machining companion grade, offers about 10 per cent greater machining speeds than Type 440C but must be heat treated with greater care to avoid cracking, (see Types 420 and 420F for data on pickling high-carbon grades).

All these high-carbon alloys have excellent edge-holding qualities and are extensively used for carving knife blades, and surgical and dental tools. They are also well known for their use in springs, valve trim, balls for grinding and bearings; oil well, needle and ball check valves; scale knife edges, spray nozzles, bottling machine parts, and seaming chucks and rolls for can-closing machines. Typical applications of Type 440C are shown in *Figs.* 18 and 19.

FERRITIC CHROMIUM TYPES (*nonhardenable by heat treatment*): The alloys in this group cannot be hardened to any appreciable extent by quenching from high temperature. They generally are used in the annealed or soft condition. These alloys serve a

variety of purposes and are available therefore in different forms. Generally, Types 405, 430 and 446 are produced in sheets, strip, plates, bars and wire; Type 406 in strip and wire only; and Types 430F and 442 in bars and wire only.

Type 405: This is a low-carbon version of Type 410 with specified aluminum content. It can be cooled from elevated temperatures without marked hardening and is used to avoid air-hardened welds.

Type 406: This is primarily an electrical resistance material. The addition of aluminum raises the specific electrical resistance and lowers the temperature coefficient. It not only serves to produce non-air-hardened welds, as in Type 405, but to increase scaling resistance as well. This grade can be hot forged or cold formed, and is easily punched, blanked or machined.

Type 430: This grade, frequently called "17 Chrome", is the most popular of the chromium grades. It possesses good mechanical and physical properties and a ductility about three-fourths that of deep-drawing quality carbon steel. In the fully annealed condition, it is particularly well suited for severe cold deformation such as in cold-heading.

Good High Temperature Properties

Because of its excellent resistance to destructive heat scaling, low coefficient of expansion, and strength at elevated temperatures, Type 430 is useful up to about 1600 F. Above this point the metal is subject to some grain growth. Consequently, welds do not possess the high degree of vibration and shock resistance of the chromium-nickel grades and welded Type 430 is unsuitable for use in rigorous service.

Long exposure to temperatures between 700 and 1100 F tends to decrease ductility and toughness when the metal is cooled to room temperatures. These properties may be restored by annealing at 1450 to 1550 F and cooling rapidly.

Applications include decorative items such as interior architectural work, automotive trim and builder's hardware. It is also used where corrosion or heat resistance, or both, are a problem, as in heat-exchanger flues, oil or gas furnace combustion chambers, wire baskets for heat treatment or pickling of parts, textile reeds and heddles, coal dewatering screens, gas burner nozzles, and stoker feedscrews.

Type 430*F:* The free machining variety of Type 430. Addition of sulfur (generally used) raises the cutting speed from 55 per cent of Bessemer screw stock speed to 85 per cent or better. Short, brittle chips are characteristic of this grade.

There is no appreciable loss in corrosion resistance. Type 430F will not discolor in ordinary atmospheres and is, therefore, widely used for products which must remain bright under relatively severe conditions, especially when hardening by heat treatment is not required. Other properties of this free cutting type are similar to those of Type 430 except that its good cold forming properties have been partially sacrificed for machinability. Some better known uses are wood screws, oil-burner nozzles, aircraft turnbuckles and various screw machine items, *Fig.* 20.

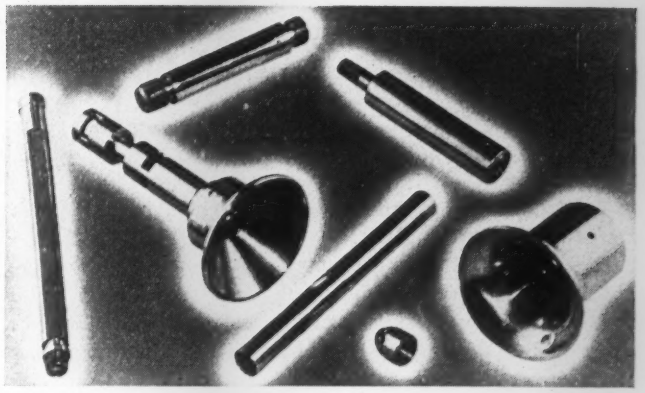


Fig. 20—Bottling machine parts including syrup dispensing tube, centering bell, nozzles, nozzle tip and cover made of free-machining stainless steel, Type 430F

Type 442: Known as "21 Chrome", this grade has heat resistance superior to that of Type 430 due to its increased chromium content. Corrosion resistance is not much greater. Its applications include use as rabble or stirring bars in the melting of nonferrous metals, shafts and pump rods in chemical apparatus and nitric acid manufacture, storage tanks, furnace parts, turbine blades, gate valves, and steam fittings.

Type 446: Commonly referred to as "27 Chrome" this grade is used almost exclusively for high-temperature scaling resistance. It resists destructive scaling at temperatures up to 2000 F in both oxidizing and reducing atmospheres. Being a ferritic chromium grade, Type 446 does not harden with heat treatment, but develops cold brittleness from heating in the 700-1100 F temperature range. Cold forming properties are only fair and machining may be done at speeds about 50 per cent of those used for Bessemer screw stock. Welds tend toward brittleness and high-temperature tensile and creep strengths are inferior to the chromium-nickel types. Uses common to this grade and Type 442 are rabble arms, glass molds, furnace parts, turbine blades, pump rods, gate valves, and steam fittings. Other applications for Type 446 include furnace and kiln linings, boiler baffles, air and gas preheaters, thermocouple wells, conveyors and solenoid valves.

Proper grade selection minimizes fabrication difficulties. However, mill processing facilities permit giving a wide range of characteristics to stainless steels for various fabricating demands. For instance, a sheet recommended for roll forming will not be the most desirable for spinning. The same mill processing that gives a sheet utmost softness by Rockwell hardness may produce a large coarse-grained structure that, when drawn, has a tendency to develop an "orange peel" surface. Changes and compromises must sometimes be made in either temper or finish. Similar considerations are necessary when dealing with bar or wire stock.

Reliable stainless suppliers know how to save manufacturers money by their material recommendations and advice. By special mill processing, a drawing operation, or perhaps an intermediate anneal and pickle, may be eliminated.

Engineer's Symposium

(*Continued from Page* 118)

or driving shaft of all loads other than torque from the eccentric units.

The sketch shows an array of power take-offs, all driven by one common high-speed shaft. The sketch may suggest possible useful application in other fields of engineering in which the writer has little or no knowledge. Its purpose is merely to show how different rotating and reciprocating speed and strokes may be taken from a common driving member.

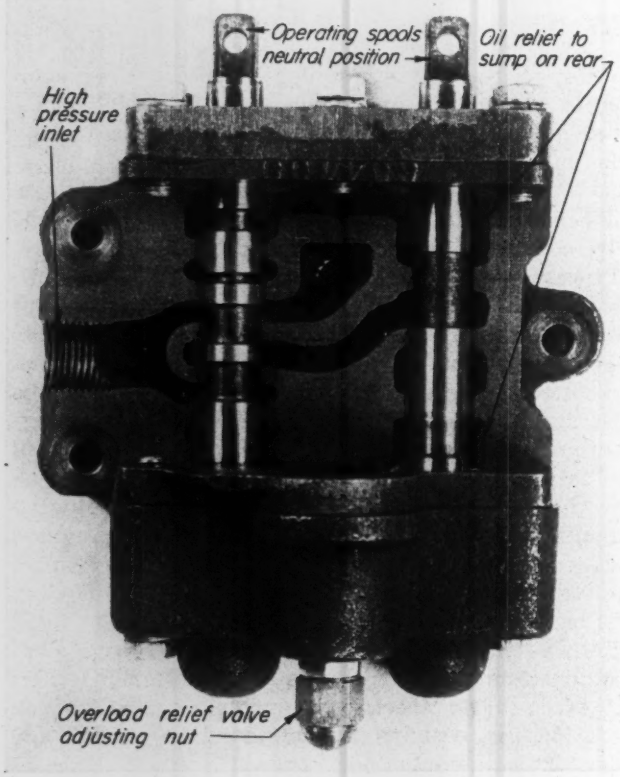
It will be noted that with two or more geared eccentric units of the same ratio, where the slow speed member of one is fed into the high speed member of another the resulting ratio is compounded. Any number of eccentric, cam, and rotating motions are possible, and may be grouped on a shaft where desired. Built as an integral unit in the form of a ball or roller bearing, they can be readily sealed and grease packed for operation in exposed locations.

Built-In Overload Protection for Control Valve

By Fred Sherriff
Mechanical Engineer
Clark Equipment Co.
Battle Creek, Mich.

OPERATION of a modern fork truck requires a dependable source of power for motions of the truck and its attachments. The Clark Carloader secures the auxiliary power requirements by means of hydraulic oil pressure piped through flexible oil lines to points of application.

Selection of a proper operating oil pressure is determined by balancing the required cross-sectional area of the operating cylinder against the ability of oil seals, hoses and pump to withstand the pressure selected. Experience indicates that the most economical balance from standpoint of design and maintenance is obtained with an operating pressure of approximately 1000 psi. Lower pressures require greater diameter cylinders and pumps with greater volumes. Higher pressures, while allowing smaller cylinder diameters, increase oil-seal maintenance, chances of oil line rupture and fitting leakage, as well as difficulties in pump design and shorter pump life.



Hydraulic pressure is maintained by a gear or vane type pump geared direct to the crankshaft of the truck engine. Oil volume is controlled by speed of the engine. On electric trucks, a similar pump is connected direct to separate series-wound electric motor which operates whenever the operator manipulates the valve control levers.

Feed to the pump is by gravity from an oil sump under atmospheric pressure. High-pressure oil from the pump is carried by hydraulic hose to multiple valves, the operating handles of which are convenient to the operator. Hoses throughout are the high-pressure, two-wire braided type, consisting of an inner synthetic tube to resist action of oil, two layers of high-tensile-strength steel-wire reinforcing, separated by elastic synthetic rubber, a layer of braided cotton cord to act as a binder of the cover stock to the hose, and an outer cover of synthetic rubber to resist weather and abrasion.

Operating valves are generally known as "self-centering spool type." The body of the valve contains oil ports which are connected to the proper oil lines through sliding spools or plungers shown in the photograph of a cut-away valve. In the neutral or self-centered position the oil from the pump passes directly through the valve body to the sump with little flow restriction. This acts as a no-load by-pass for the pump when pressure is not required and allows the pump gears or vanes to turn freely without

waste of energy. It must be remembered that on gasoline models the pump is turning whenever the truck engine is operating.

When the spool is raised or depressed by the operator, the no-load by-pass to the sump is closed and oil is then directed under full pressure to the desired cylinder. Since no by-pass or overload valve is provided in the pump itself, an overload by-pass is contained in the valve body to relieve oil to the sump whenever pressure exceeds 1200 psi. This will occur at the end of the stroke of the piston in the cylinder and no damage will result in applying full pressure to a cylinder after full travel has been obtained. When applying pressure to one side of the piston of a hydraulic cylinder it is necessary to relieve the oil trapped on the opposite side of the piston. This is accomplished within the valve itself by allowing the spool to open a return port to the sump. Depressing the spool allows high-pressure oil to enter one end of the cylinder, trapped oil to leave the other. Raising the spool above neutral reverses this operation. When the spool is in the neutral or self-centered position, the ports both to and from the operating cylinder are closed, which locks the piston in the desired position. Banks of valves may be used allowing installation of any number of operating cylinders.

Hydraulic cylinders are honed to a microinch finish to insure leak-proof seating of the piston leathers throughout the entire stroke of the piston. Oil in the hydraulic system is independent of the truck engine oil system and has low-foaming characteristics and a viscosity approximately that of SAE 10.

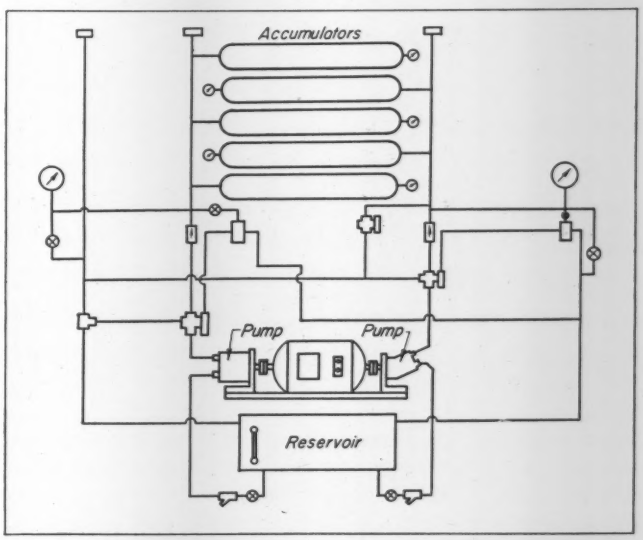
Hydraulic System Designed as Package

By Julius Kendall
Greer Hydraulics Inc.
Brooklyn, N. Y.

HITHERTO, the use of hydraulics has been more or less limited because, in addition to designing a machine, it was also necessary to design a complete hydraulic system for its operation. The latter, however, has been eliminated by the recent introduction of the "Power Pack" illustrated. It is furnished as a completely self-contained and compact hydraulic system including all necessary components such as pumps, motors, reservoir, valves, manifolds, filters, controls, etc.

A feature of this unit is the use of hydraulic accumulators in the circuit. Especially designed for operation with a molding press, the unit can deliver 124 gpm flow at 2000 psi pressure during the short time required by the press. To obtain these conditions without the use of accumulators would require an electric motor with an output of 145 hp and a large-volume pump. However, by incorporating five 10-gallon hydraulic accumulators in the circuit, the manufacturer was able to reduce the size of the motor to 30 hp with a resultant saving in original cost and in upkeep of the system. This is made possible by the fact that the circuit is of the intermittent-duty type.

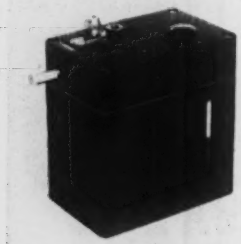
As shown in the accompanying schematic diagram, the Power Pack is a dual-circuit unit. It incorporates a 200-psi low-pressure circuit and a 2000-psi high-pressure circuit. A steel cabinet houses all the circuit components. Removable panels are provided for ease and maintenance. The hydraulic circuit is powered by a dual-shaft, 30 hp electric motor which is suitably protected by an overload cutout magnetic starter and is controlled by a start-stop pushbutton. One end of the motor shaft drives a low-pressure pump capable of delivering 58.5 gpm; the other end drives a high-pressure pump with a capacity of 19 gpm. Low-pressure fluid is stored in three 10-gallon accumulators and is released instantly when the system calls for it. High-pressure fluid is stored in two 10-gallon accumulators. The low-pressure accumulators operate between 200 and 300 psi, while the high-pressure accumulators operate between 2000 and 3000 psi. Other components of the system are: Relief and control valves, 100-gallon reservoir equipped with filters, low-pressure gage and high-pressure gage. Each accumulator is equipped with a charging valve and an air-pressure gage for use in preloading the accumulators and for indicating at a glance the preload pressure at any given time. The circuit is equipped with a two-inch low-pressure line, a two-inch high-pressure line, and a three-inch return line.



new parts and materials

To obtain additional information on these new developments see Page 331

Automatic Lubricating Unit



Designed for installation within small machines, the type M lubricator can oil up to 100 bearings simultaneously and with measured regularity. Automatically driven, by the machine in which it is installed, this cyclic unit develops an average discharge pressure of 40 psi and discharges from 2.5 to 5.5 cu cm at each pump impulse. The lubricator is available in several reduction ratios permitting the selection of the correct rates to accommodate the drive speed and the time cycle desired. Also, it is made in both lefthand and righthand drives. Pump unit of the lubricator may be mounted in a built-in reservoir integral with the machine to be lubricated or it may be furnished complete with a standard reservoir, provided with an oil sight gage. Housing measures 3 inches wide, 4½ inches long and 5 inches high; tapped mounting holes are provided in base. Manufacturer: Bijur Lubricating Corp., 43-01 22nd St., Long Island City 1, N. Y.

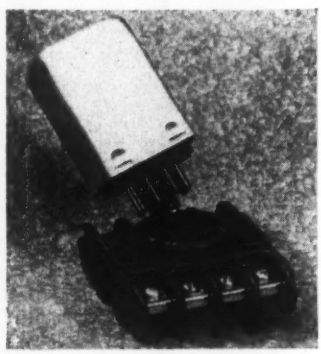
For further information circle MD 1 on card Page 331

Fast-Cycling Pneumatic Valve

Electrically operated 4-way valve for directional control of the flow of compressed air is designed to provide fast-acting pushbutton control for small air-operated units. Valve also may be used as an automatic control for any air-operated unit through the use of limit switches. It is a packless, reciprocating disk type, pilot-operated and solenoid controlled. The piston is aluminum and the disk is hardened stainless steel. Built in two types, Type I is arranged for double solenoid operation using two switches, valve position being determined by energizing the proper solenoid. Type II is arranged for power-offset operation using a single switch. Valve returns to its initial position when the solenoid is de-energized. Either type is capable of as many as 180 operating cycles per minute in handling air pressure from a minimum of 25 psi to a maximum of 150 psi. Type I units are rated 10 watts; type II, 20 watts. Standard models are built for 115-volt, 50-60 cycle operation. However, they can also be supplied for 230-volt power or other frequencies or direct-current use. Three styles are available in both types. NE-37 has a ⅜-inch female inlet and cylinder, and ½-inch exhaust connections; NE-50 has ½-inch inlet and cylinder and ¾-inch exhaust, while Model NE-75 has ¾-inch inlet and cylinder and 1-inch exhaust. Manufacturer: Hannifin Corp., 1101 S. Kilbourne Ave., Chicago 24.

For further information circle MD 2 on card Page 331

Totally Enclosed Relay

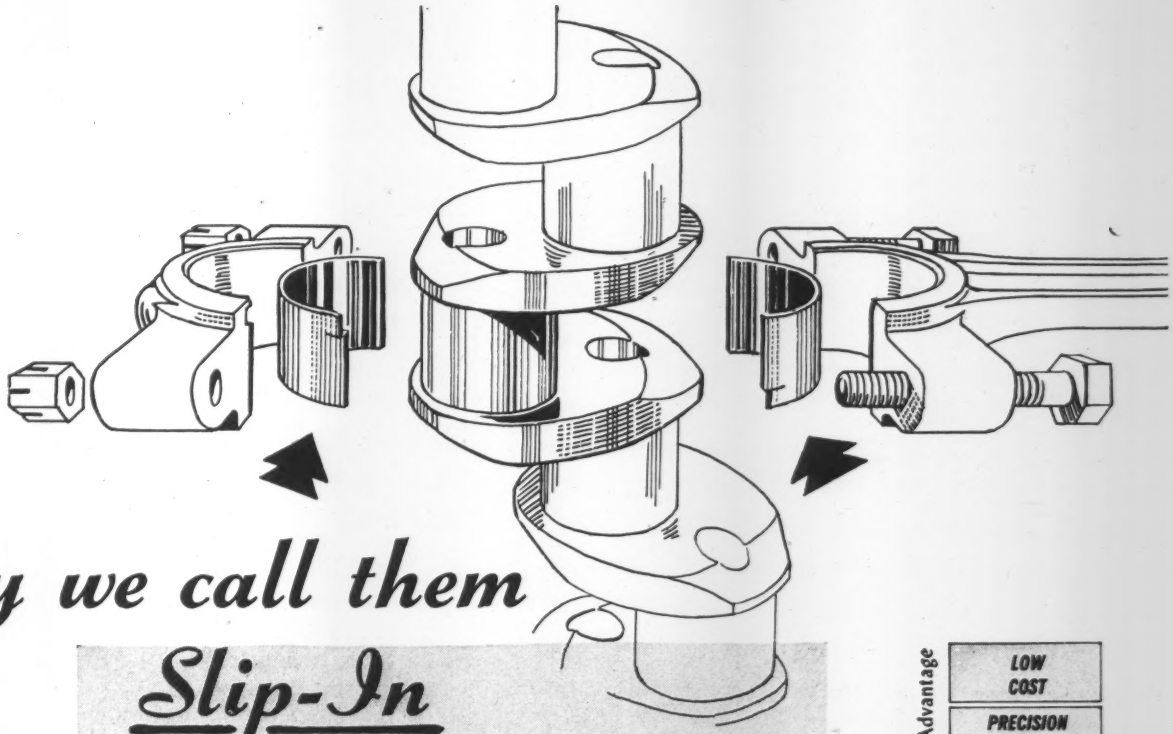


Relay for use with such low-load devices as mercury thermostats is totally enclosed and operates on only 2 milliamperes with 115-volts ac across the coil circuit. Series 41 relay is available in several styles, typical one illustrated weighing 3 oz, and measuring 1¼ by 1¼ by 2 inches. Manufacturer: Sigma Instruments, Inc., 70 Ceylon St., Boston 21.

For further information circle MD 3 on card Page 331

Compact Gearmotors

Built in six sizes and capable of delivering from one to sixty hp, new gearmotor line consists of units having a speed range of 7½ to 780 rpm, using single, double or triple-reduction types. Since flange-mounted motors are used, many standard motor enclosure types are readily adaptable to the reduction unit; all enclosure types can be bolted to the gear elements without alteration. The



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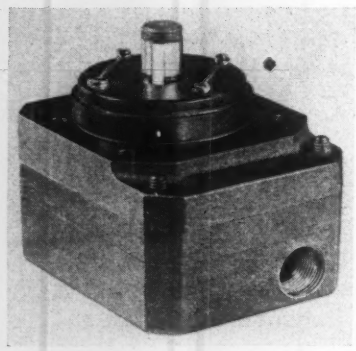
single reduction unit has a cast-iron housing, bolted to and supported by the motor frame. Output shaft is located centrally and splash lubrication is provided for the helical pinion and gear. In the double-reduction type, feet are integral with the gear housing, a footless motor being utilized; the three-stage gearmotor is made by adding a one-stage reduction assembly bracket to the two-stage gearmotor. The entire line of reducers have each pinion and gear well supported and spaced in order to permit high load-carrying capacity. Antifriction bearings are used throughout. The units have been designed for heavy duty and have a high power-volume ratio. Output shaft has been centrally located to simplify application and facilitate use in tight spots. Manufacturers: Reliance Electric & Engineering Co., 1079 Ivanhoe Road, Cleveland 10; and Philadelphia Gear Works, Inc., Erie Ave., and G St., Philadelphia 34.

For further information circle MD 4 on card Page 331

Hydraulic Gear Pumps



Two new hydraulic gear pumps, models 9000 and 9075, have been developed for continuous duty at pressures up to 1000 psi and intermittent service up to 1500 psi. Model 9000 has a capacity of 8 gpm at 1800 rpm; model 9075 has 15-gpm displacement at 1200 rpm. Units are suitable for pressure lubrication or oil-hydraulic service in systems ranging from 100 to 1500 psi. Utilizing a minimum number of moving parts, pumps have Nitralloy gears operating inside chrome-plated faces of cast-iron housing. Manufacturer: Aro Equipment Corp., Bryan, Ohio.

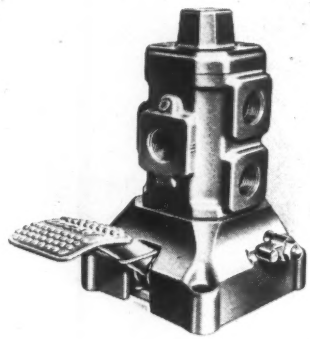
For further information circle MD 5 on card Page 331

Fractional-Horsepower Capacitor Motor

Small and powerful capacitor type motor, particularly applicable where space limitation is a prime consideration, is known as the Raytheon 340. It is a distributed winding unit and can be designed to match the requirements of any application. Available for clockwise, counterclockwise or reversible two or four-pole operation, the motors have starting torques as high as 50 oz-in. and are made in ratings from 1/50 to 1/25-hp. Two styles of housings—steel or die-cast zinc—are in production. The die-cast housings are available in open or totally enclosed styles for horizontal or vertical operation; both models have wick-fed Oilite bearings with large reservoirs and oil slingers as well as internal cooling fans. Manufacturer: Russell Electric Co., 340 W. Huron St., Chicago.

For further information circle MD 6 on card Page 331

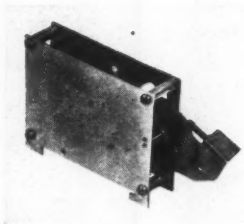
Foot-Operated Hydraulic Valve



Free or latching control of double-acting hydraulic cylinders is provided by 4-way hydraulic valve rated 1500 psi. Known as model 7025, it is a balanced-piston unit of the sliding piston type. It has no valve seats or packings other than the stem seals at each end which are subject only to exhaust pressure and is made in standard port sizes from ½ to 1 inch. Special units for operating pressures to 2500 psi in oil circuits and 1500 psi in water service, can also be obtained. Manufacturer: Logansport Machine Co., Logansport, Ind.

For further information circle MD 7 on card Page 331

Mechanical Control Timer

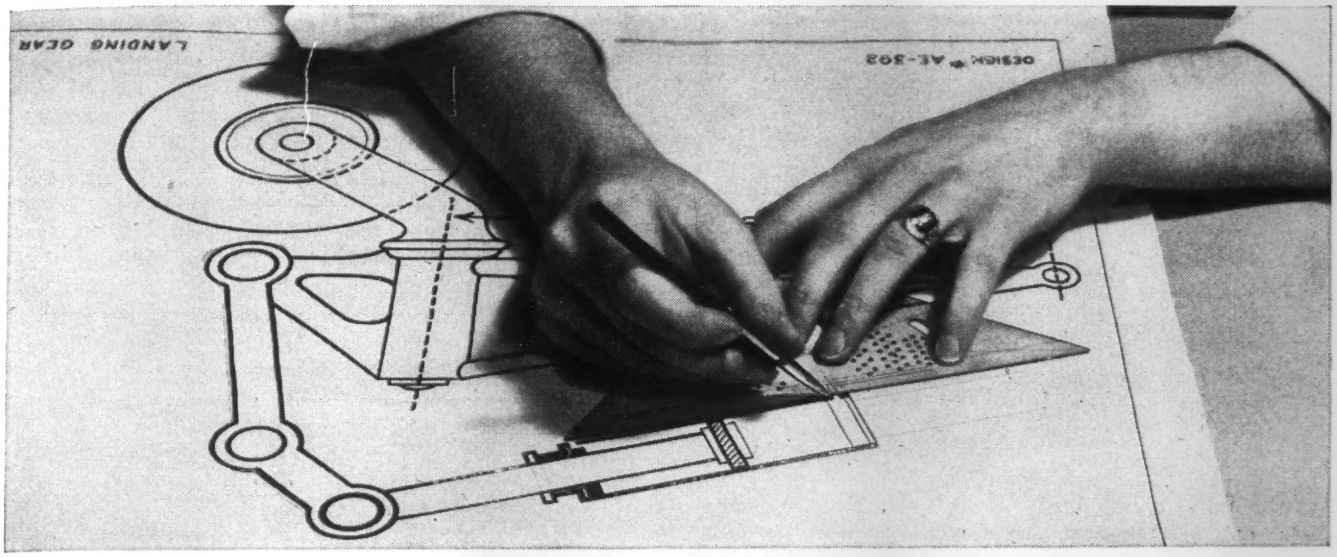


Control instrument for use in measuring small time intervals is made in either ratchet or friction drive models. The unit measures periods from 5 to 30 minutes, has dimensions of 1¾ by 2½ by ⅝-inch. Features of the timer are smooth, quiet, nonpositional operation, and compact construction. Newly developed balance and escapement assembly results in efficient timekeeping under variable spring loading. Manufacturer: Mechanical Industries Production Co., 217 Ash St., Akron 3.

For further information circle MD 8 on card Page 331

Quick-Connecting Steam Coupling

Handling pressures up to 350 psi, new steam coupling may be quickly connected or disconnected. Connection is made by pushing the plug into a socket where it remains locked until released by movement of the socket sleeve. Two styles are made, low pressure and high. High-pressure type incorporates a socket and nipple that act as a gland on a replaceable, formed type of packing. It is designed for pres-



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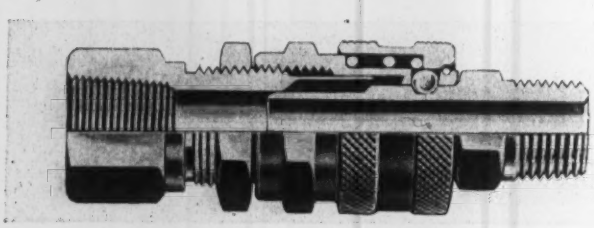
sures from 40 to 350 psi and is made in two sizes: Series 2500 have 5/16-inch bore and Series 4500 have ½-inch bore. Low pressure types are designed for operation at 0 to 40 psi, a replaceable, steam washer



type packing being used. Sizes available are Series 2000 with ¼-inch bore and Series 4000 with ½-inch bore. Thread sizes in each range from ⅛ to ⅝-inch. Series 4000 and 4500, however, are made with ½-inch pipe threads only. Manufacturer: Hansen Mfg. Co., 4031 W. 150th St., Cleveland 11.

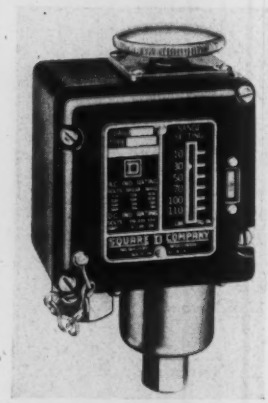
For further information circle MD 9 on card Page 331

Electron-Tube Socket

Features of the new Bakelite tube socket include high-conductivity contacts, above or below horizontal panel mounting, screw type terminals and long surface creepage distance for high-voltage applications. Design is based upon the electrical and structural requirements of all tubes using the RMA Super Jumbo tube base, as well as industrial 411 and 412 bases which have the same pin arrangement and smaller shell diameter. Manufacturer: American Phenolic Corp., 1830 S. 54th Ave., Chicago 50.

For further information circle MD 10 on card Page 331

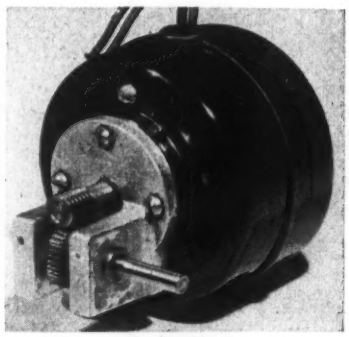
Machine-Tool Pressure-Actuated Switch



Pressure-type switch for use in controlling hydraulically or pneumatically actuated machines has an operating rate ranging up to 300 cycles per minute. Known as Class 9012, this control unit is of the positive quick make-and-break type having a single-pole, double-throw mechanism. Separated, nonoverlapping, normally-open and normally-closed circuits with double-break silver contacts are provided. Two basic types, ACW and ADW, are available. Bellows-actuated models are made in a variety of ranges up to 1000 psi capacity. Piston-actuated controls, for use with hydraulic circuits, are suitable for pressures up to 3000 psi. In use, a trip indicator operates behind a window in the cover, permitting observation of switch action. The indicator flag facilitates proper setting of the trip point and indicates any pressure surge or other condition that might be objectionable. Die-cast, the aluminum enclosures are drip-tight and oil resistant; exterior mounting lugs are not required. Manufacturer: Industrial controller division, Square D Co., 4041 N. Richards St., Milwaukee 12.

For further information circle MD 11 on card Page 331

Shaded-Pole Gearmotor

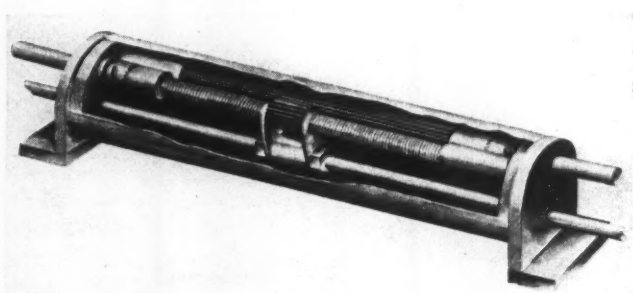


Fractional-horsepower shaded-pole induction motor is rated 1/100-hp, 110 volts, 60 cycles. The four-pole motor has worm reduction unit mounted outside motor housing; gears are hobbed from alloy steel and mounting bracket is cast aluminum. The motor housing is extra-heavy-gage steel, available in enclosed or ventilated types. Features of the unit include interlocking shock-proof construction, large oil reservoirs and vacuum-impregnated windings. Manufacturer: E. L. Klauber Machinery Co., 3221 Olive St., St. Louis 3.

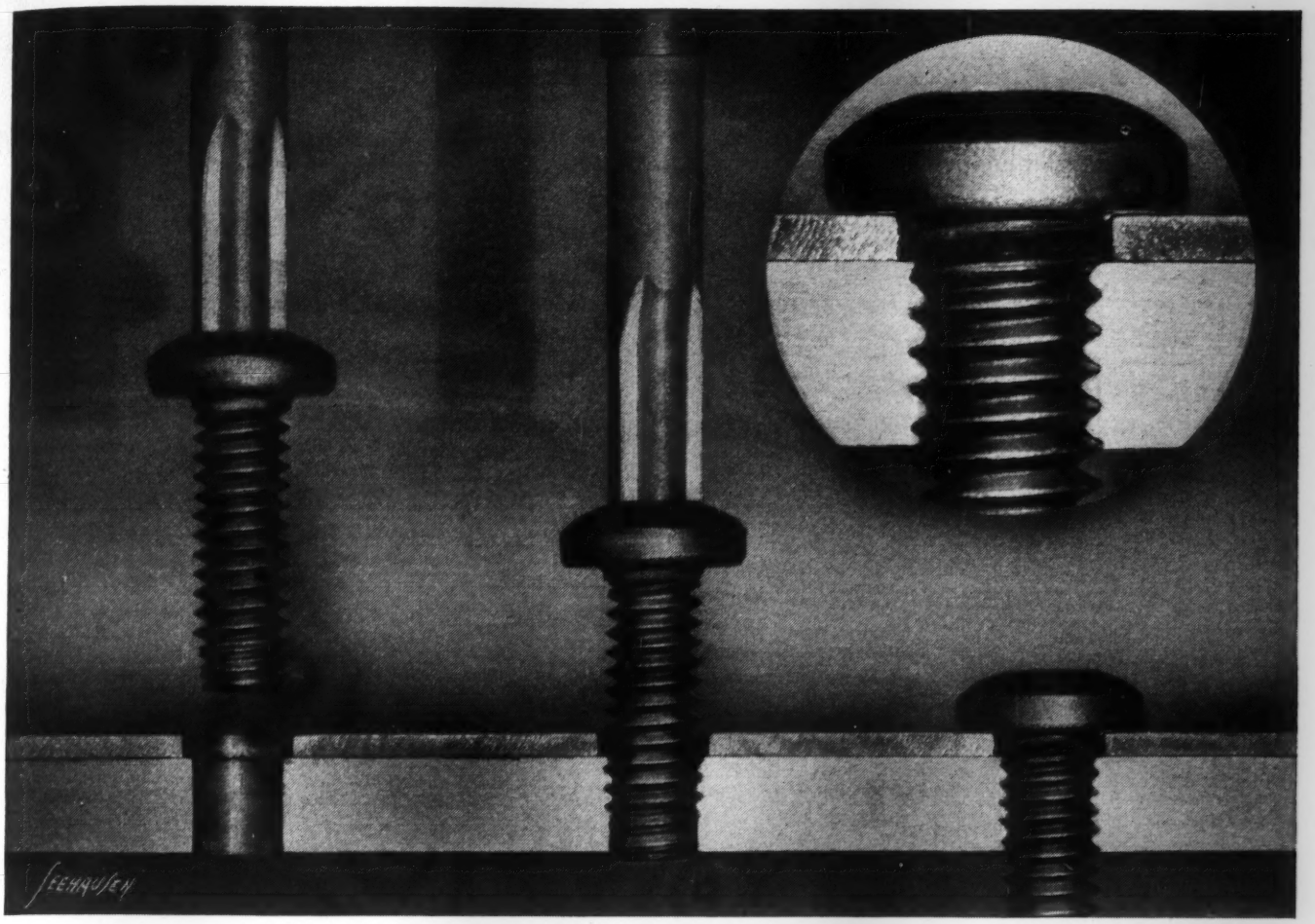
For further information circle MD 12 on card Page 331

Power Synchronizer

Designed to provide absolute synchronization between two sources of power, new synchronizer weighs 3 lb and measures 10½ inches long. Unit functions through the use of gear trains, lengthwise main pinion being attached to the main source of power. Second power source drives lengthwise screw on which rides



a gear meshing with the main pinion. If both pinion and screw are rotating at the same speed, there will be no lateral motion of the gear. If, however, there should be a speed differential, a lateral motion



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3 ***Thus a 3-Way Saving***—elimination of separate tapping operation; cost of multiple fastenings; and cost of slow-down in two-handed assembly.

4 ***The Design of the Clutch Recess*** is "a natural" to meet the rigor of the toughest thread-forming job.

5 ***Note the Husky Teeth*** of the straight-walled recess . . . to stand up under the highest torque required.

6 ***Also Consider*** the rugged structure of the straight-sided Type "A" Bit for longer uninterrupted service.

7 ***It Is Important*** to the safety and speed of assembly operation that the CLUTCH HEAD engagement is *definitely non-tapered.*

8 ***Hence, There Is No "Ride-out"*** to combat . . . eliminating both the skid hazard and need for fatiguing end pressure.

9 ***Freedom*** from burred or chewed-up heads also speeds assembly. The Center Pivot column guides driver into dead-center entry for automatic straight driving.

10 ***Note How the Lock-On,*** exclusively a CLUTCH HEAD feature, permits of one-handed driving. A simple reverse turn of the screw in the recess locks screw and bit as a unit.

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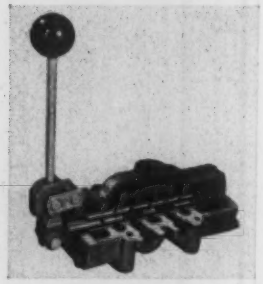
will be imparted to the gear. This motion is transferred to the governor or speed control through linkages. Unit is recommended for control of internal combustion engines as well as electric motors. Manufacturer: Taber Associates, 60 E. 42nd St., New York City.

For further information circle MD 13 on card Page 331

Hydraulic Selector Valves

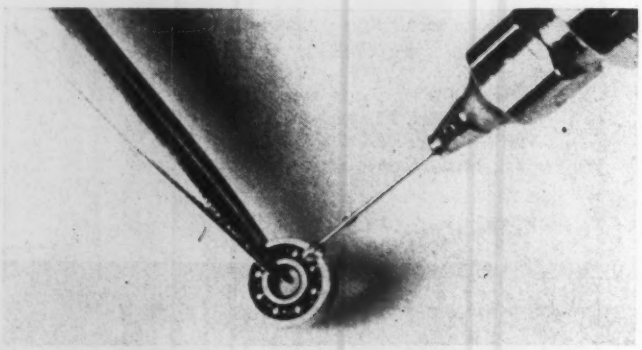


Series of hydraulic 4-way selector valves are designed for 1000-psi service in the control of one double-acting cylinder or two single-acting cylinders. Spool-type construction permits a wide variety of porting arrangements for either open-center or closed-center systems. Feature of the valve is the completely hydraulically balanced spool. It is balanced axially as well as radially so that there is no possibility of spool movement caused by surge pressures in the return line; large diameter of the spool assures maximum oil flow with minimum pressure drop. To fit particular requirements, valves are available having spring-centered, spring offset, or three-position detent action. Units are also made with interchangeable end caps to facilitate placing the operating lever at either end and at any one of four radial positions. Models available include: Lever operated for manual control, mechanically operated for remote manual control, pilot operated, single treadle for foot control, solenoid operated for remote electrical control and Camroller operated for automatic control. Sizes include ¼, ⅜, ½, ¾ and 1-inch ports with capacities up to 32 gpm. Manufacturer: Adel Precision Products Corp., Burbank, Calif.

For further information circle MD 14 on card Page 331

Miniature Nonmagnetic Ball Bearing

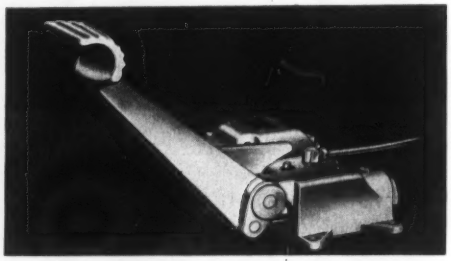
Developed for electrical, electronic or other applications requiring freedom from magnetic influences, new ball bearing has ¼-inch outside diameter, 5/64-inch bore and 3/32-inch width. Bearing material is beryllium copper, precipitation hardened to 42 Rockwell C. Tensile strength of the material at this hardness is about 200,000 psi, and the material is noncorrosive. Bearings are offered in both Class 1 and Class 5 precision tolerances with Class 7 also available on order. They are of full-race design for maximum capacity. Manufacturer: New Hampshire Ball Bearings, Inc., 5 Micro Circle, Peterborough 1, N. H.



For further information circle MD 15 on card Page 331

Hydraulic Foot Pump

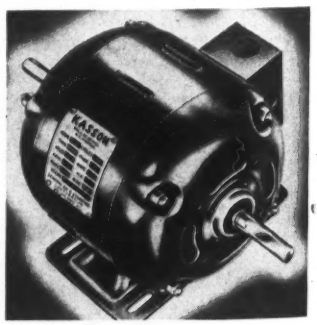
Compact, lightweight, foot pump weighing only 13½ lb, will develop pressures up to 1500 psi. The unit is designed so that the pedal-return spring is inside the pump body and the release controlled by slightly raising the single pump pedal. The 15 cu in. of hydraulic oil which is used is contained in a sealed reservoir requiring no vent. Pump can therefore be



mounted in offset positions to suit requirements. Features include built-in oil strainer, adjustable relief valve and a chrome-plated piston for smooth, long-life operation. Manufacturer: Lyon-Raymond Corp., 3840 Madison St., Greene, N. Y.

For further information circle MD 16 on card Page 331

Fractional-Horsepower Motors

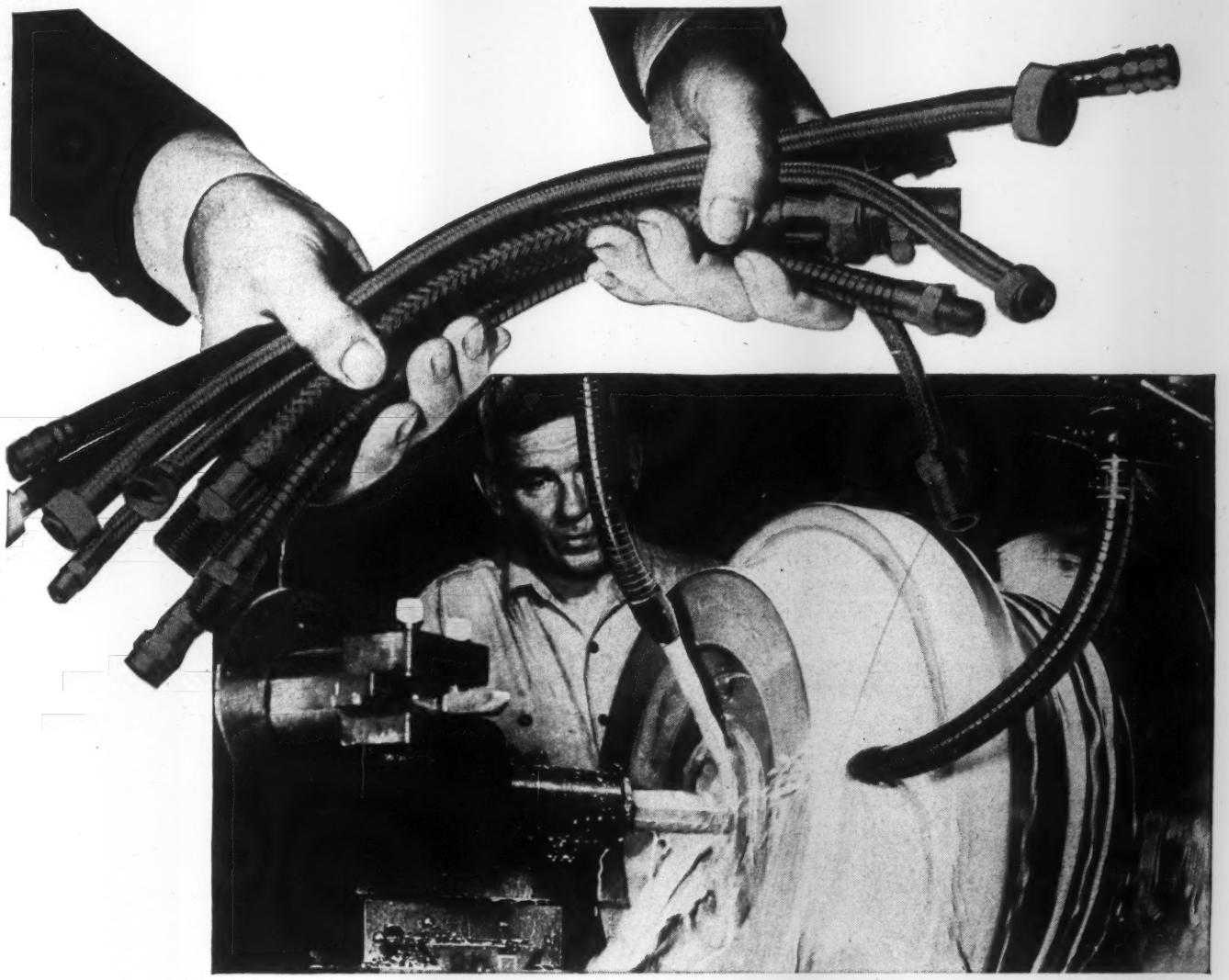


All-steel clad 110-volt, 60-cycle motors are available as 1/3-hp units in split phase, 3-phase or capacitor start types or ½-hp units, 3-phase only. Particular features of the induction motors are: Pressed-steel end bells, dual-flow ventilation, sealed ball bearings and standard double-end shafts. Manufacturer: General Die and Stamping Co., 262-272 Mott St., New York 12.

For further information circle MD 17 on card Page 331

Three-Way Pneumatic Valves

Designed for use with single-acting air cylinders, the 1291-N3 valves are recommended where it is desired to accurately control and hold any given cylin-



Conveying coolant through American Oil Feed Tubing. This tubing, of oil-tempered steel, is ideal for such use because it will stay in any position into which it is bent.

PHOTO COURTESY WARNER & SWASEY CO.

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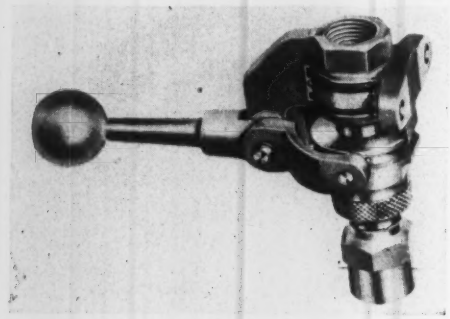
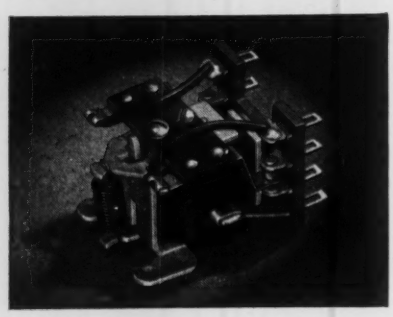
der position. Units are 3-way, neutral-position, hand-operated valves having stainless-steel body, brass sleeve and stainless-steel adapter. They are fitted with a forged bronze head having a flat spring that



indexes and holds the handle in on, off, hold or neutral position. Models are available in ⅜ and ½-inch pipe sizes for use with air pressures up to 250 psi. Manufacturer: C. B. Hunt & Son, Inc., Salem, Ohio.

For further information circle MD 18 on card Page 331

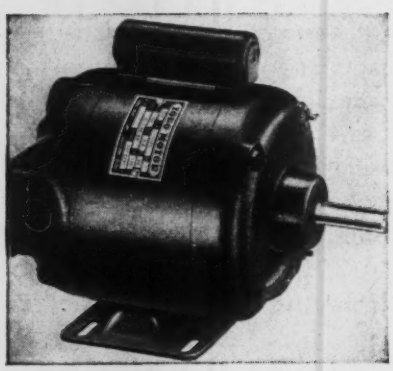
Miniature Double-Pole Relay



Double - pole, double - throw relay is designed for electric vending machines and similar equipment where there is need for small, low - cost relays having easy mounting and simple wiring arrangements. This type 119XBX relay is rated five amperes and is available in models for operation at any a-c voltage up to 115. Manufacturer: Struthers-Dunn, Inc., 150 N. 13th St., Philadelphia 7.

For further information circle MD 19 on card Page 331

Fractional-Horsepower Electric Motors



Line of capacitor start, fractional - horsepower motors are available in 1/6, ¼, 1/3 and ½-hp ratings. They have high starting torques being particularly adaptable for drives such as pumps, compressors, stokers, etc. The units, which are known as type CSI, are of splash-proof design with sturdy steel frames and die-cast aluminum-alloy end bells having cast-in terminals for easy wiring. Bases are solid or resilient and are slotted for easy belt adjustment. Either 115 or 230-volt single-phase 60-cycle styles are available. Manufacturer: Torq Electric Mfg. Corp., 1075 Interstate St., Bedford, Ohio.

For further information circle MD 20 on card Page 331

High-Vacuum Solenoid Valve

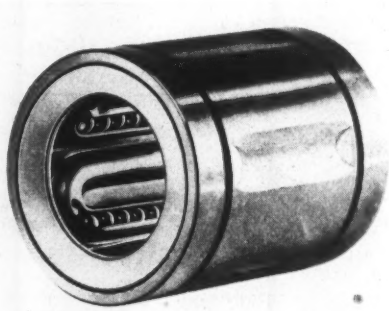
Solenoid-operated high-vacuum valve requires no pilot or other medium for operation. Unit is of poppet design with corrosion-resistant internal parts.

Resilient synthetic-rubber seat provides a positive seal that makes the valve leakproof. Unit operates on 110-volt d-c power. Manufacturer: Airmatic Valve Inc., 1643 E. 40th St., Cleveland.

For further information circle MD 21 on card Page 331

Ball Bushing

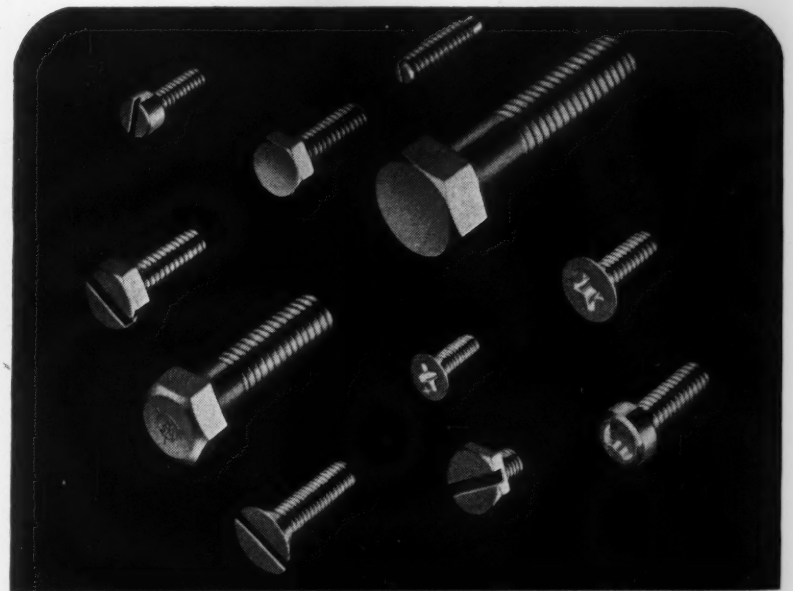
Ball-type bearing, designed on the circulating-ball principle, is suitable for reciprocating or sliding motion as well as rotation. The new unit, known as the series A ball bushing, is available for shafts in diameters of ¼, ½, ¾ or 1 inch with other sizes soon to be placed in production. The bearing consists of three basic parts in addition to the balls. The first is a sleeve made of "ball-bearing" steel, hardened and ground to close tolerances. Second is a pressed-steel

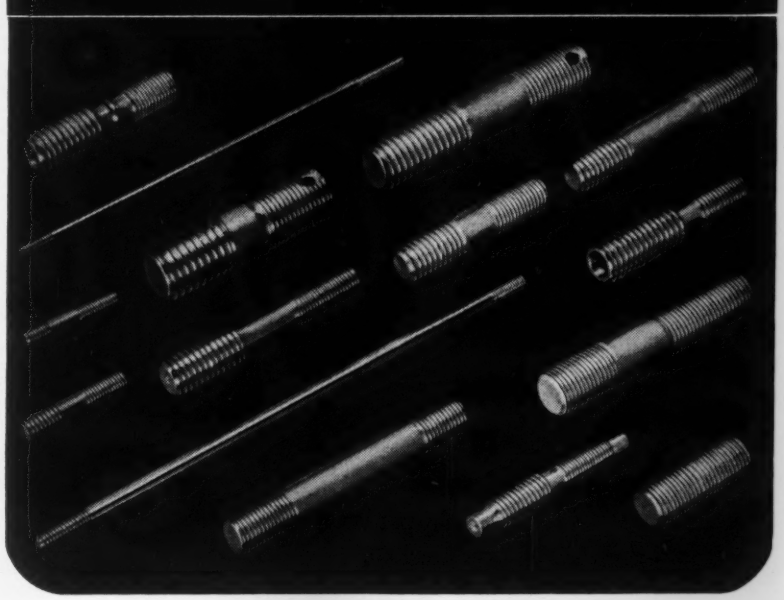


retainer which fits inside the sleeve. Third is a pair of rings which are pressed in the ends of the sleeve to position and secure the retainer. The bearing is said to have a low friction coefficient, preventing

The uses of "LOK-THRED" are unlimited

STUDS, BOLTS OR SCREWS LOCK MORE SECURELY...BECOME *TIGHTER* IN SERVICE...FORM A SEAL THAT LIQUIDS UNDER PRESSURE CANNOT PENETRATE





The revolutionary "LOK-THRED" that has proved so successful in stud applications can now be made in any type of threaded fastener.

Now the uses of "LOK-THRED" are unlimited. Whether you need a tiny screw or a heavy-duty bolt, you can have the headed fastener you want—threaded with "LOK-THRED".

With the positive sealing action of "LOK-THRED", bosses and blind tapping can be eliminated in many cases. Air and liquids are held under pressure without leakage—suggesting many new uses, such as for water pumps or for cylinder heads.

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Look at all these Advantages

1. Lock securely and become tighter in service.
2. Have much higher fatigue limits than studs with conventional threads.
3. Stronger in both tension and torsion than ordinary American National Threads.
4. Carry entire normal working load on 6° angle at root of thread under high compressive prestress.
5. Modified American National Threads permit use of standard tools.
6. Re-usable and on any reapplication less than one-half additional turn brings torque back to its original installation value.
7. Do not require selective fits.
8. Do not gall when being driven or fret in service.
9. Act as dowels and taper pins.
10. Seal positively and eliminate added bosses and blind tapping.

THE NATIONAL SCREW & MFG. COMPANY, CLEVELAND 4, OHIO

cocking and binding; its free-rolling action maintains precise alignment by the elimination of wear. Manufacturer: Thomson Industries, Inc., 1029 Plandome Rd., Manhasset, N. Y.

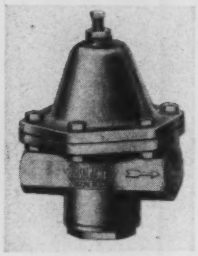
For further information circle MD 22 on card Page 331

Liquid Adhesive

Two modified phenolic liquid adhesives designated G-E adhesives Nos. 12507 and 12508 are designed for cementing metals, thermosetting plastics, wood or fabric to themselves or each other. The new bonding agents have high shear and tensile strength and excellent resistance to water, gasoline, kerosene and mineral oils. They are said to be especially useful for bonding metals and laminates and attaching brackets and lugs to thin metal sheets. Adhesive No. 12507 has high solvent resistance. It is intended for applications which require an effective adhesive with a high shear strength, and requires facilities for prebaking the adhesive film before pressing. Adhesive No. 12508 is recommended for applications in which the adhesive must possess exceptionally high cleavage strength. Higher application pressures are required for assembly. Manufacturer: General Electric Co., Pittsfield, Mass.

For further information circle MD 23 on card Page 331

Reducing and Regulating Valve



Automatic pressure reducing and regulating valve is suitable for use with pressures up to 250 psi. Known as type E-41, the valve is of all-bronze construction and is made in pipe sizes of 1¼, 1½ and 2 inches. Special features include yoke construction, Neoprene seats and Hycar washers. Manufacturer: A. W. Cash Valve Mfg. Co., 6613 E. Wabash Ave., Decatur, Ill.

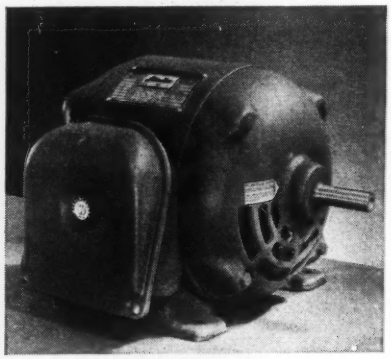
For further information circle MD 24 on card Page 331

Constant-Volume Gear Pump

Made in eight different capacities ranging from 3 to 10 gpm at 1200 rpm, the Model 5W pump is particularly suitable for materials-handling equipment, heating units and similar installations requiring a low-cost, quiet and efficient pump. A constant-volume pump for 1000 psi service, it features the Rota-Roll design. In this design, the smaller roller, or inner member, is keyed to a shaft and drives the outer member at a speed 25 per cent lower than the speed of the motor. Efficiency of the unit is about 85 per cent and the pumping members have a special, sturdy tooth form which provides long pump life. Model 5W is suitable for direct coupling only. Other models can be furnished for sheave or pulley drive. Manufacturer: Hydraulic division, Sundstrand Machine Tool Co., Rockford, Ill.

For further information circle MD 25 on card Page 331

Single-Phase Induction Motor



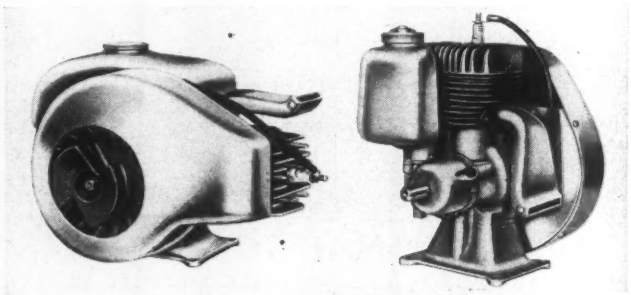
Single - phase, capacitor - start, induction - run, squirrel-cage motor is made in seven ratings: ¾, 1, 1½, 2, 3, 5 and 7½ hp. It is of all-steel construction in frame sizes 203 through 324. Designated type CAP, it utilizes the capacitor for starting only, a switch disconnecting the capacitor as the motor approaches full speed; it then operates entirely as an induction motor. Models are available for 60, 50 and 25-cycle, single-phase, 110-220 dual voltage or single voltage operation. Speed ratings made include 3450, 1750 and 1160 rpm at 60 cycles; 2875, 1475 and 970 rpm at 50 cycles, and 1450 rpm at 25 cycles. Manufacturer: Westinghouse Electric Corp., P. O. Box 868, Pittsburgh 30, Pa.

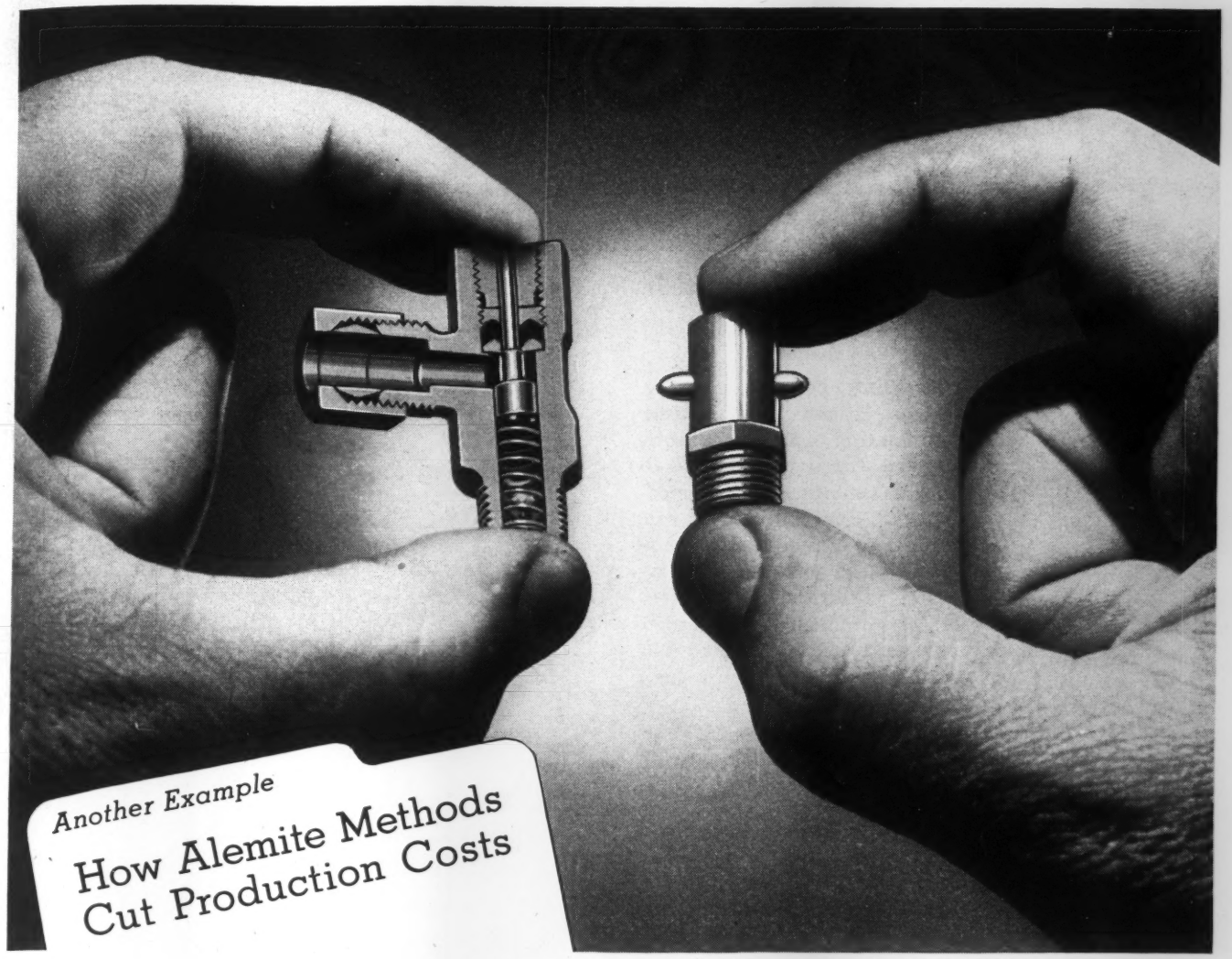
For further information circle MD 26 on card Page 331

Single-Cylinder Gasoline Engines

Made in two styles, the Emco single-cylinder, two-cycle engines develop ¾-hp at 2400 rpm. Model 2CH-1000 and model 2CV-1000 are both made of weight-saving materials and have 1⅝ in. bore and 1⅝-inch stroke. They are available in both vertical and horizontal-mounting styles and are designed for general applications such as powering lawnmowers, pumps, portable tools, washing machines etc. Manufacturer: Erie Engine & Mfg. Co., 995 E. 12th St., Erie, Pa.

For further information circle MD 27 on card Page 331





"Midget" Centralized Lubrication System Helps Machines Produce More

A LOW COST SYSTEM THAT'S HARDLY LARGER THAN AN ALEMITE PIN-TYPE FITTING

Protect Those Machines You Design from Human Error

Human error in lubrication can easily destroy the efficiency of the best designed machine. A five minute conference with an Alemite Representative may well reveal lubrication design features that will make your machines produce more. This man is fully qualified to help you because he is a specialist in modern methods of handling and applying lubricants. Write to Alemite, 1804 Diversey Parkway, Chicago 14, Illinois.

Small, Simple, Compact . . . this new Alemite "Midget" measuring lubrication valve is designed for all types of heavy, light and precision machines — especially where space limitations are encountered. With this single line system, valves mounted on or near the bearings deliver a measured amount of lubricant from one safe, central point while machines keep right on producing!

Alemite Centralized Lubrication Systems make possible the lubrication of hundreds of bearings from one central point—greatly reduce the possibility of "human error". There is no danger of over-lubrication or under-lubrication . . . handles grease or oil. Oilers work in complete safety. Result: Production goes up . . . costs go down.

Just one of many ways that Alemite Methods help industry add productive time to machines with faster, more efficient lubrication equipment and procedures. The Alemite Representative can show case after case where a time study analysis has proved that his methods reduce costs. These methods eliminate costly, time-consuming handling of oils and greases . . . slash "time-out" for lubrication . . . keep dirt and moisture out of lubricants . . . completely mechanize lubrication from barrel-to-bearing. These benefits all add up to lower production costs and longer machinery life. Get the complete facts, today.

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Another Product of STEWART WARNER

MODERN LUBRICATION METHODS THAT CUT PRODUCTION COSTS

engineering dept equipment

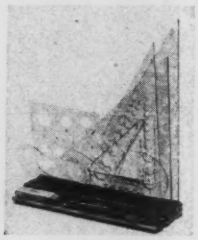
In order to obtain additional information on this new equipment see Page 331

Electric Erasing Machine

Hollow-shaft erasing machine has 7-inch erasers, saving time and eliminating waste. Erasers are inserted through either end of hollow shaft and secured by simple chuck. Especially designed induction motor mounted in rubber assures smooth, quiet running and vibration-free service. Motor is provided with sealed ball bearings and, having no brushes, is nonradio interfering. Manufacturer: Charles Bruning Co. Inc., 4754 Montrose Ave., Chicago 41.

For further information circle MD 28 on card Page 331

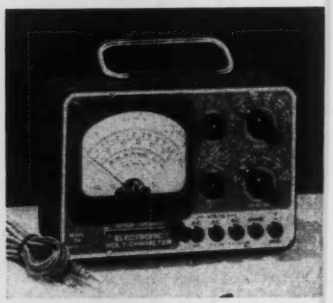
Drafting-Tool Tray



Combination of a rack and tray, Draftsman's Tool tray will accommodate triangles, rulers, pens, pencils and drawing instruments in easily accessible positions. Curves and triangles are held upright where they can be easily identified while pencils and similar instruments are held in rack for easy grasping. The new tray is metal and weighs 8 oz; it measures 3⅝ inches wide, and 8 inches long. Manufacturer: Winslow Product Engineering Corp., 614 S. Maple Ave., Los Angeles 14.

For further information circle MD 29 on card Page 331

Wide-Range Electronic Volt-Ohmeter



Self-contained battery-operated volt-ohmeter known as model 214 is designed for applications requiring resistance and voltage measurements at high impedance. Meter is electronically protected against burnouts and requires no warm-up period. Ranges covered are: 0-1200 volts ac and dc in six ranges; 1 ohm to 1000 megohms in six ranges; 10 mmf to 15 megohms a-c input impedance, and 15 megohms d-c imput impedance. Instrument weighs 5¼ lb and measures 8¼ by 6 by 4¼ inches. Manufacturer: Hickok Electrical Instrument Co., 10545 DuPont Ave., Cleveland 8.

For further information circle MD 30 on card Page 331

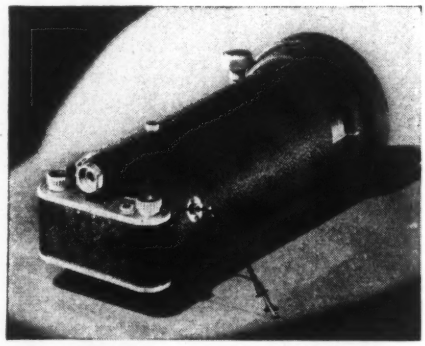
Steel Drafting Stool

Featuring a revolving, spring-filled all metal seat covered in leather or leatherette, Royal No. 524 drafting stool is of welded construction. Screw stem is tool steel and its housing its especially designed with small threads to prevent wobble, adding to the life of the stool. Seat height is adjustable from 22 inches to 28 inches, or 27 inches to 33 inches. Metal portions of the stool are finished in durable taupe-color oven-baked enamel. Manufacturer: Royal Metal Mfg. Co., 175 N. Michigan Ave., Chicago 1.

For further information circle MD 31 on card Page 331

Oscilloscope Camera

Clamping on to the supporting ring of the cathode-ray tube of any standard 5-inch oscilloscope, type 271-A oscilloscope camera is automatically positioned

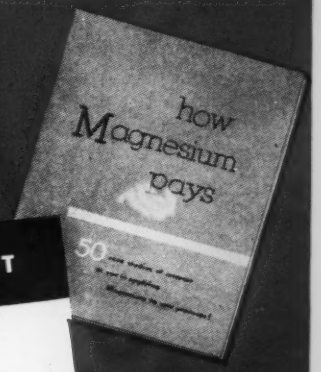


for correct focus. Adjustable mounting permits the camera to be horizontal, vertical or tilted for corresponding images. Unit is a 35 mm camera with fixed-focus f:3.5 coated lens and having simplified shutter with time, bulb and 1/30-second exposures. Screen image is observed through a peep-hole at the camera end of the light hood and exposure is made by cable release. Manufacturer: Allen B. DuMont Laboratories, Inc., 2 Main Ave., Passaic, N. J.

For further information circle MD 32 on card Page 331

Take a leaf from this revealing **new book!**

Here's the inside story on how manufacturers of a wide range of products—from parts for aircraft to portable tools—have found that it pays to use magnesium. It's a book of actual case studies of successful magnesium applications—a book that will be of direct interest to you.



TRANSPORTATION EQUIPMENT

REFRIGERATED TRUCK BODY

PROBLEM

Increasing operating costs are a serious problem to fleet operators. This is particularly so with refrigerator trucks, which have a low ratio of payload to gross weight, and have to make frequent stops and starts. Both of these factors were involved in the operations of a Michigan meat packer, whose distribution system involves the use of refrigerator trucks with salesmen—drivers for selling direct from the trucks to the meat retailers. This fleet operator decided that the higher cost of operation could best be lowered by significantly reducing the weight of his equipment.

To achieve a maximum weight saving, the operator decided to use magnesium to construct the refrigerator truck body. Anticipated savings in operating costs justified a higher initial investment in this equipment, but actually an unexpected saving was realized when it was found that the use of magnesium would save 3,000 pounds in the body weight, permitting the unit to be mounted on a 1½ ton chassis instead of the 2½ ton chassis required for the steel body—and without any reduction in carrying capacity. Gas consumption, tire wear, and maintenance costs have all been reduced beyond expectation. Progressive operators all over the country are finding that light weight magnesium bodies are an effective means of cutting costs.

SOLUTION

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St. Louis • Houston • San Francisco • Los Angeles • Seattle
Dow Chemical of Canada, Limited, Toronto, Ontario

Pneumatic Cylinders

(*Concluded from Page* 125)

These indicator cards together with the load-stroke curves frequently are of material aid in improving cylinder control.

CONTROL VALVES FOR PNEUMATIC CYLINDERS: Normally an air cylinder requires pressure to move either way so the control valve must drain one end of the cylinder to exhaust at the same time line pressure is being admitted to the opposite end. This requires one four-way valve or two three-way valves linked together. *Fig.* 13 shows schematically what the valve must do. A drum type four-way valve is shown although it may be one of several types.

There are such a myriad of different air valves for this type of service that it is impossible to consider more than the general types. *Fig.* 14 shows a few examples of the more common types of valves.

Poppet Valves: These valves may have hardened seat or may use resilient material against a hardened surface. They may be unbalanced so that the pressure forces cause high seating loads and high opening forces or they may be balanced to reduce these values. Poppet valves are characterized by a definite parting of the valve and seat when opened.

Sliding Valves Simple in Design

Sliding Valves: These valves may be rotary or linear in movement and are characterized by hard, smooth sliding surfaces. They are difficult to balance, so the pressure forces cause high friction forces. Further, they generally are not absolutely air tight unless the surfaces are lapped to optical flatness. Sliding valves have the advantage, however, that three and four-way porting can be accomplished much more simply than in other types.

Sliding Valves with Seals: It is possible to design valves that retain the same general scheme of operation as the sliding type valve except that sealing is effected by some resilient material rather than the close sliding fit. With this valve it is necessary to seat or unseat the soft sealing material under pressure so that operation is limited to pressures that will not damage the soft seat under these conditions.

Valves Using Flexible Tubes or Diaphragms: The life and operating pressures of these valves are limited, but generally they are simple, inexpensive and in some types very lightweight.

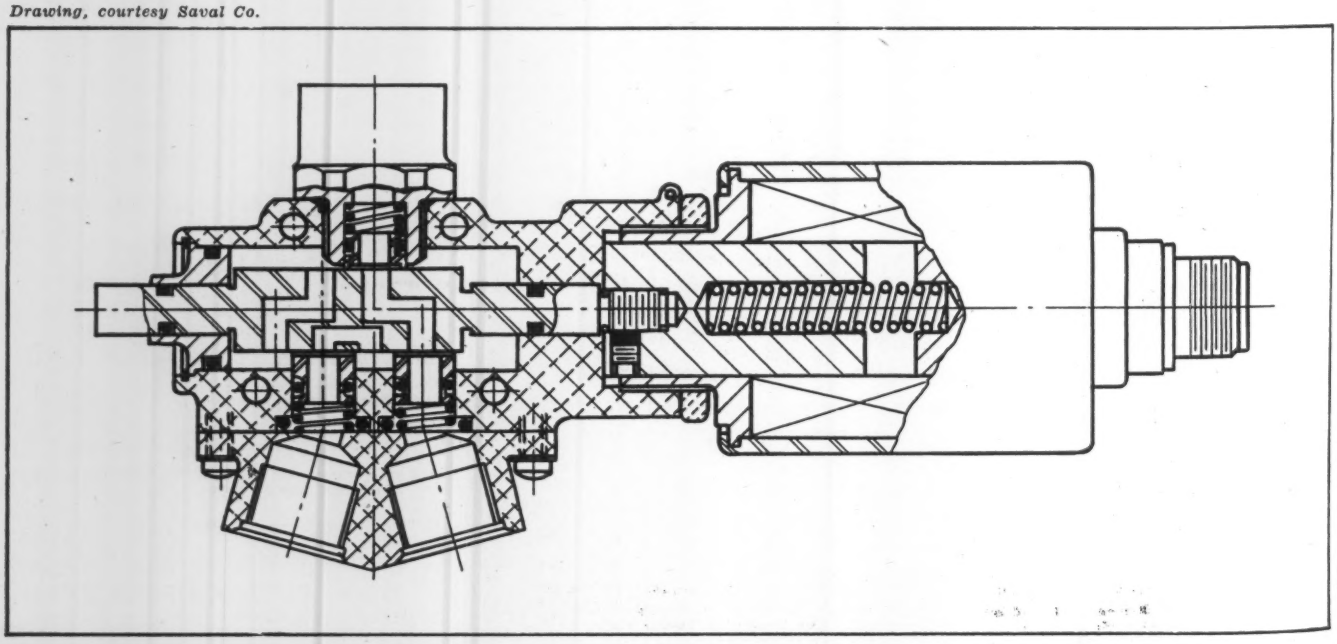
Variety of Control Methods Possible

Control valves for air cylinders may be operated manually, electrically, mechanically, hydraulically, or even automatically, i.e., from the piston rod itself. This last case may be illustrated in a cylinder where piston movement is initiated from some outside source allowing the piston to travel a full stroke, where it automatically moves the valve to the reverse position, causing the piston to return and stay until operation is again initiated. *Fig.* 15 shows an aircraft type solenoid-operated, four-way valve with sliding lapped seals for use on high-pressure air. The solenoid for this service has two windings, a large one for a large "pull-in" force and a small one for "holding". The large winding is cut out by a small switch when the valve has completed most of its travel.

Since most control systems have to be particularly suited to the application, only general principles and typical examples have been cited in the hope that they may point the way toward more competent analyses of specific design problems.

Fig. 15—Lightweight, solenoid-operated sliding type four-way valve for cylinder control on 1500-psi air pressure. Valve utilizes pressure-loaded metallic sealing element sliding on optically flat surfaces and incorporates manual override for emergency operation

Drawing, courtesy Saval Co.



• It's called a saw, but it really doesn't saw . . . it cuts railroad rails, I-beams, steel bars and heavy pipe in two by melting the metal. Whether it is high carbon or alloy steel, hardened or annealed, the contact saw does the job by creating sufficient heat-generating friction between saw blade and steel to actually melt the metal. The saw blade teeth simply wipe away the molten metal. And the V-Belts that drive the blade must be able to deliver positive power at high, sustained speeds. They must be able to withstand the shock that occurs when saw blade first contacts the metal to be cut. And they must operate on extremely short centers. The manufacturer found these characteristics in Dayton Steel Cable V-Belts and standardized on them.

The contact saw is another example of the ability of Dayton V-Belt Drives to provide positive power transmission under difficult conditions. This ability to operate in excess of standard requirements under all operating conditions . . . unaffected by oil, heat, dust or liquids . . . is one of many reasons why more and more industrial designers consistently specify Dayton V-Belts for original equipment. A Dayton power transmission specialist is ready to help you. For additional information write: *The Dayton Rubber Company, Dayton, Ohio.*

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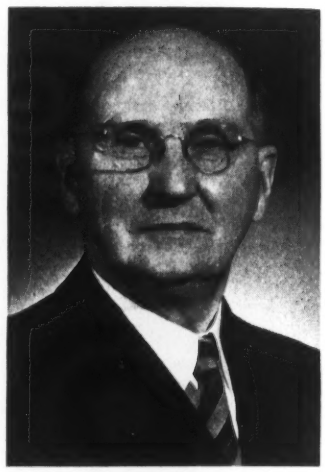


Hans P. Dahlstrand

HANS P. DAHLSTRAND recently has been appointed director of engineering of the steam turbine department, Allis-Chalmers Co. Mr. Dahlstrand, who came to the United States in 1903, is a graduate of Boros College of Technology, Boros, Sweden, and has been associated with Allis-Chalmers since 1904 when he accepted a position as a designing engineer with the company's then newly organized steam turbine department. In 1908 he was named engineer in charge of all detail design for that department and in 1915 became engineer-in-charge of turbine design. He was appointed consulting engineer in 1942. Mr. Dahlstrand is a fellow of the American Society of Mechanical Engineers and has been granted about twenty patents covering steam turbine developments.

Concurrently, E. H. HORSTMAN was appointed chief engineer and is in complete charge of steam turbine engineering.

A. E. W. Johnson

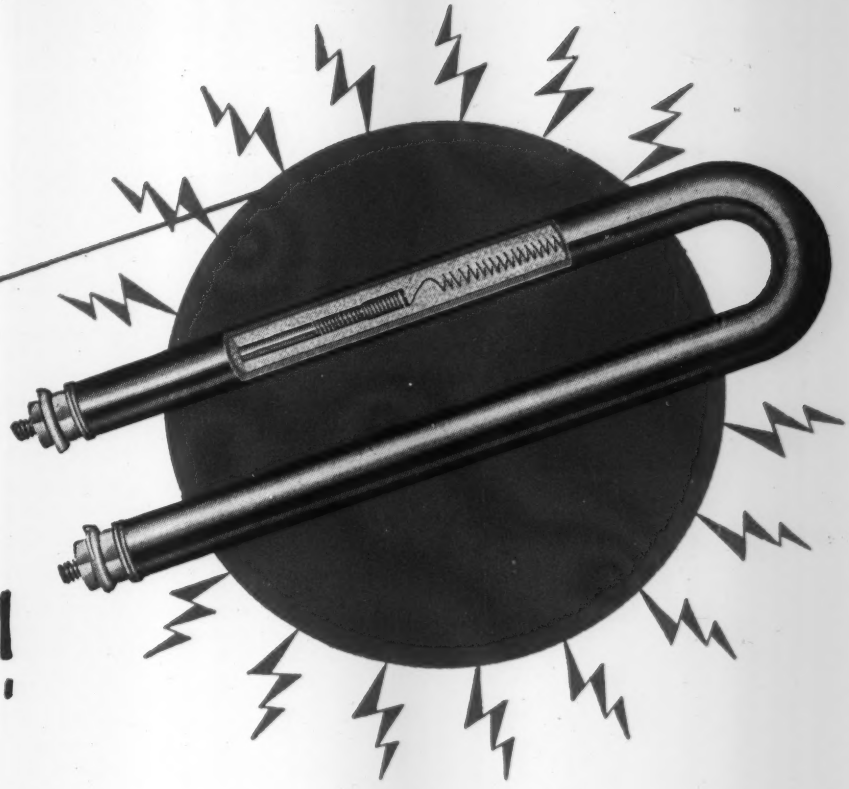
A. E. W. JOHNSON, for thirty-one years engaged in engineering and development work for the International Harvester Co., recently was named director of engineering for the entire company. Mr. Johnson began in 1916 as a draftsman in International's McCormick Works, Chicago, where he assisted in the development work on the early Farmall tractor and other power-driven machines. Advancing through various engineering assignments, Mr. Johnson was appointed manager of implement engineering in 1940, and when the company divisionalized its operations in 1944, he became manager of engineering for the farm implement division.

J. Harry DuBois

J. HARRY DUBOIS, executive engineer of the Shaw Insulator Co., has been elected president of the Society of Plastics Engineers at its annual meeting held recently in Detroit. Mr. DuBois served as the society's vice president last year in addition to being eastern regional chairman and a member of its publications and membership committees. He became active in the plastics industry immediately following graduation from the University of Minnesota, joining the General Electric Co. in 1927, serving with that organization until he became associated with Shaw Insulator in 1944. In his present position as executive engineer he is responsible for engineering, research and sales. As an authority on plastics, Mr. DuBois is the author of two books used extensively in the industry, and has served as technical consultant to the U. S. Office of Education.

JULES MULLER has been appointed director of engineering of the E. W. Bliss Co., manufacturer of presses, rolling mills and can machinery. Associated with the company since 1945, Mr. Muller had, prior to his recent

YALE Hottest News In Electric Heat!



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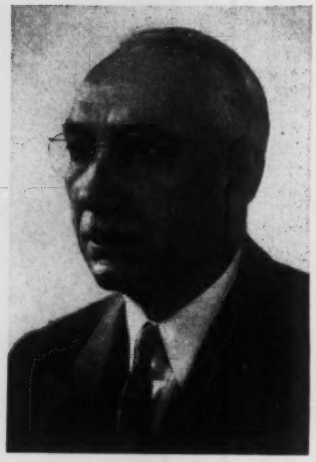

Jules Muller

appointment, supervised welding design and manufacture for its machinery. A graduate of the University of Illinois, Mr. Muller has also served with A. O. Smith Corp., Henry Voght, and Leeds & Northrop, and for a time was an engineering consultant to the welding industry. Arthur Schloz continues as chief engineer of the Press division and H. Farrington as chief engineer of the Rolling Mill division.

T. L. SHERMAN has joined Fairbanks, Morse & Co. as consultant designer to be associated with the company's recently organized engine design section.

T. L. Sherman

Following graduation as a mechanical engineer from the University of Queensland, Brisbane, Australia, Mr. Sherman was engaged in the design of internal combustion engines of all types for companies in Australia, England, France and Germany. He came to the United States in 1929 to take charge of the development of Michell-Crankless engines, particularly those of the two-cycle, opposed-piston type. In addition he has served as consultant to the Sterling Engine Co., Buffalo, on its two-cycle engines and during the war undertook the development of a slant-type, opposed piston diesel aircraft engine for the Navy Department.

♦

A. J. LANGHAMMER, president, Amplex Div., Chrysler Corp., has been named recipient of the Stevens Institute award for outstanding achievement in the development and use of powder metallurgy in the production of bearings and machine parts. Mr. Langhammer has been president of Amplex division since 1937 and has written numerous papers and handbook data on tool steels and various phases of powder metallurgy.

♦

FRANK R. DEBARTOLO has been promoted from chief engineer to assistant to the vice president, Lombard Corp., and will be in charge of all development and promotional activities for the company. Succeeding him as chief engineer is ALFRED W. SCHULTZ, formerly with the Warren City Mfg. Co. EDGAR P. HULSE, for the past several years associated with C. H. Hunt Inc., has been named assistant chief engineer.

♦

PETER ALTMAN has been appointed vice president, Continental Motors Inc., Detroit, and will be in charge of its newly organized Multi-Tool division. For the past five years Mr. Altman has served as consultant and chairman of the company's engineering council.

♦

T. R. WIESEMAN, chief executive engineer of the Louis Allis Co., has been elected to the company's newly created office of vice president in charge of engineering. With the organization for twenty-two years, Mr. Wieseman has been closely identified with its engineering developments. JAMES H. DAGANHARDT has been promoted to chief engineer from his previous position as chief supervisory engineer.

♦

HANS BOHUSLAV has become associated with Engineering Controls Inc., as vice president in charge of engineering. For the past five years Mr. Bohuslav held a similar position with the Sterling Engine Co., previous to which he had been with the Enterprise Engine & Foundry Co.

♦

H. G. ERICKSON has been made chief engineer of the Luscombe Airplane Corp., succeeding E. W. NORRIS. Mr. Erickson previously had been chief structural engineer for the company.

♦

DR. JOHN A. HUTCHESON has been appointed director of the Westinghouse Research Laboratories, succeeding DR. L. WARRINGTON CHUBB. Dr. Chubb, retiring for reasons of health, has been named director emeritus and will continue in an advisory capacity. Dr. Hutcheson, for the past four years associate director of the laboratories, directed the company's extensive wartime radar research program and later the formulation of plans for atomic energy development.

♦

DR. HALDON A. LEEDY, chairman of physics research, Armour Research Foundation of Illinois Institute of Technology, has been named acting director of the foundation. He succeeds DR. JESSE E. HOBSON, who resigned recently to accept directorship of the Stanford Research Institute.

♦

CLARENCE J. SMITH recently has been appointed chief engineer of the Hydro-Line Mfg. Co., manufacturer of hydraulic and air cylinders and special machinery. Mr. Smith, an engineering graduate of Pratt Institute, has previously served as chief engineer in charge of new hydraulic adaptations for the John S. Barnes Corp., and more recently as general manager of the Process Equipment division of W. F. & John Barnes Co.

♦

DR. GALEN B. SCHUBAUER and DR. HAROLD K. SKRAMSTAD, aerodynamicists of the National Bureau of Standards, have been presented the Reed award of the Institute of the Aeronautical Sciences "for their contributions to the understanding of the mechanism of transition from laminar to turbulent

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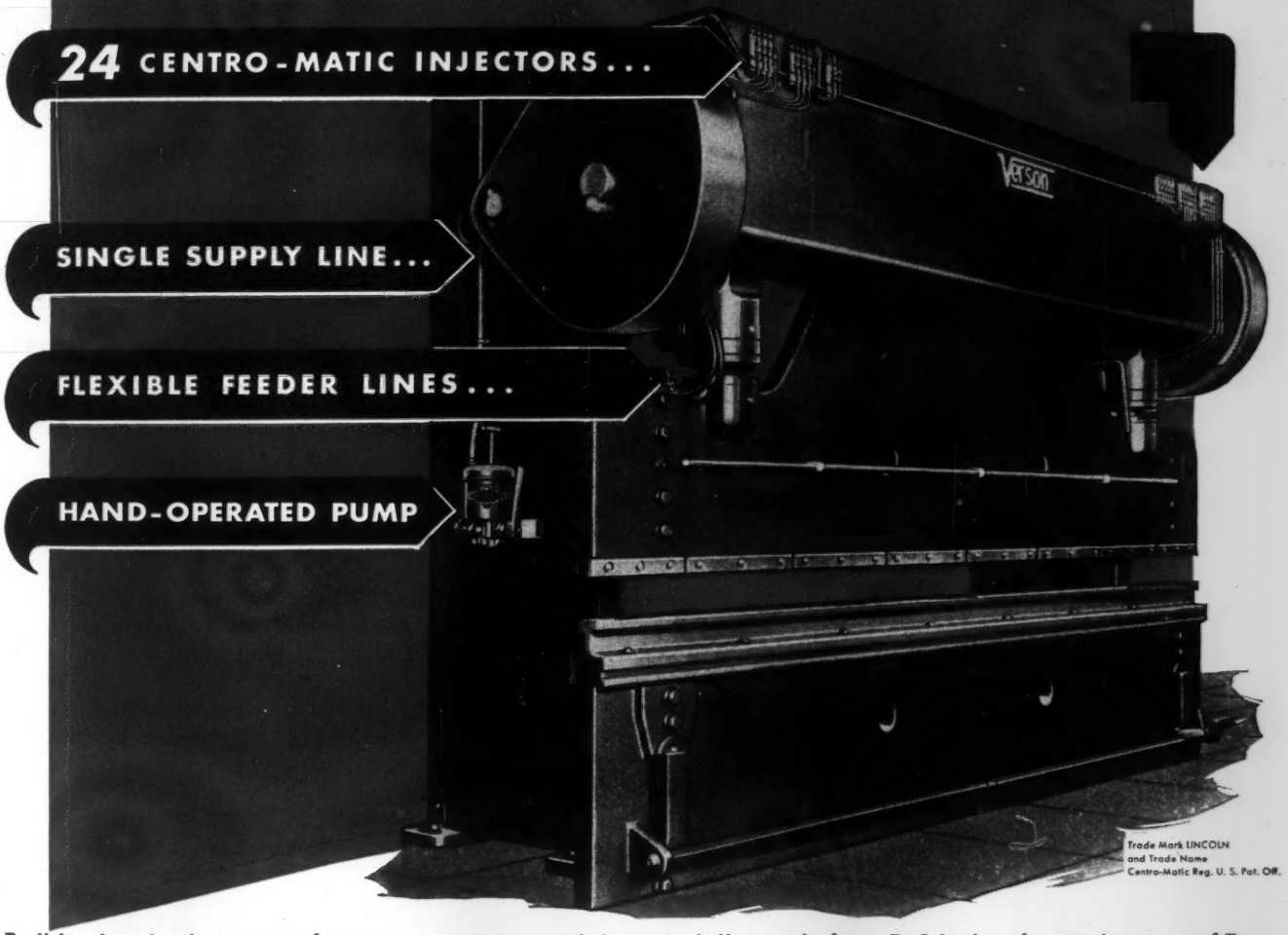
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flow." Dr. Schubauer is chief of the Bureau's aerodynamics section and Dr. Skramstad, chief, guided missiles section.

♦

LEONARD A. STEWART has accepted the position of chief engineer, American Coach & Body Co., Cleveland. For the past eight years he had been a body engineer in the truck division of Mack Trucks, Inc.

♦

WILLIAM F. BELL has been appointed director and vice president in charge of engineering and production of Southern Electric Inc., Hammond, Ind.

♦

ALBERT R. HUTCHINGS recently was named executive engineer to be in charge of engineering and manufacturing for the Carlyle Johnson Machine Co., Manchester, Conn. His previous associations include positions with Kenneth A. McIntyre Associates, B. F. Perkins & Son Inc., and Compo Shoe Machinery Corp.

♦

EDGAR H. KRAINER has been appointed chief engineer of the U. S. Broach Co., replacing RANDALL L. KAMISCHKE.

♦

JOSEPH J. HITE has been promoted to the post of assistant chief engineer, American Hoist & Derrick Co. Mr. Hite formerly held the position of development engineer and more recently that of office engineer.

♦

C. B. BRADISH has been named manager of engineering of the recently established control division of the apparatus department, General Electric Co.

♦

ROBERT H. JEWETT, formerly chief of preliminary design, has been named chief project engineer, pilotless aircraft, by Boeing Airplane Co.

♦

ROBERT I. DICK has been appointed chief experimental engineer, Le Roi Co., Milwaukee, having previously held the position of chief engineer, Murphy Diesel Co., Milwaukee.

♦

E. D. WILKIN recently has been appointed chief engineer, Johnson Lawn Mower Corp., Ottumwa, Iowa. He previously had been associated with Salsbury Motors Inc., Pomona, Calif.

♦

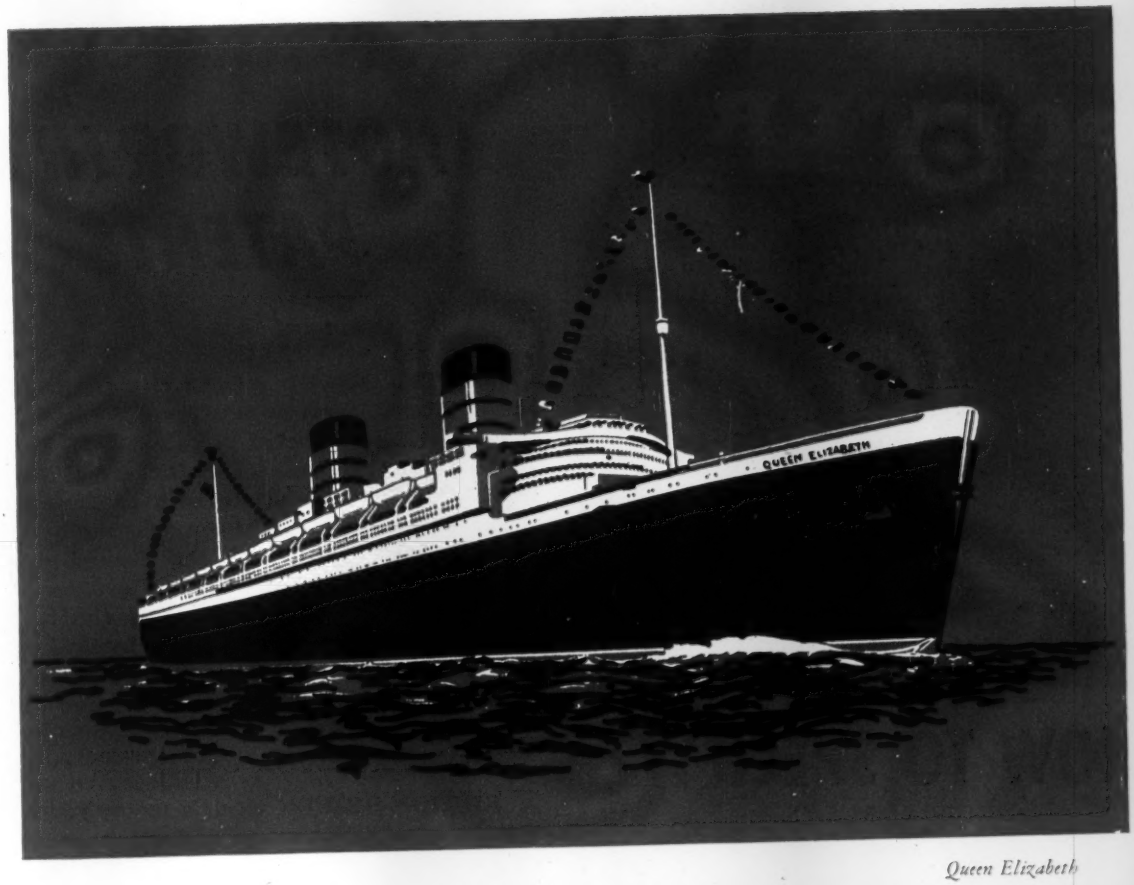
DR. WALLACE R. BRODE has been appointed associate director of the National Bureau of Standards. He succeeds DR. HUGH L. DRYDEN who recently was appointed director of aeronautical research, National Advisory Committee for Aeronautics. Dr. Brode comes to the Bureau from Ohio State University where he was professor of chemistry.

♦

MATTHEW E. HAMILTON, formerly a project engineer with Harry Ferguson Inc., is now chief engineer for the Stockton Works, J. I. Case Co., Stockton, Calif.

♦

DONALD L. HERR has become associated with the engineering department of Allen-Bradley Co., to develop servomechanisms for machine tool and other motor controls.



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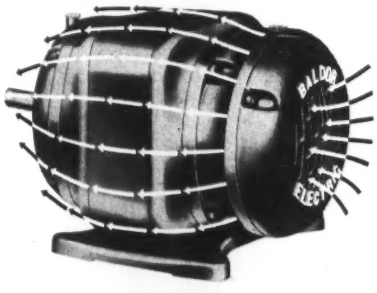

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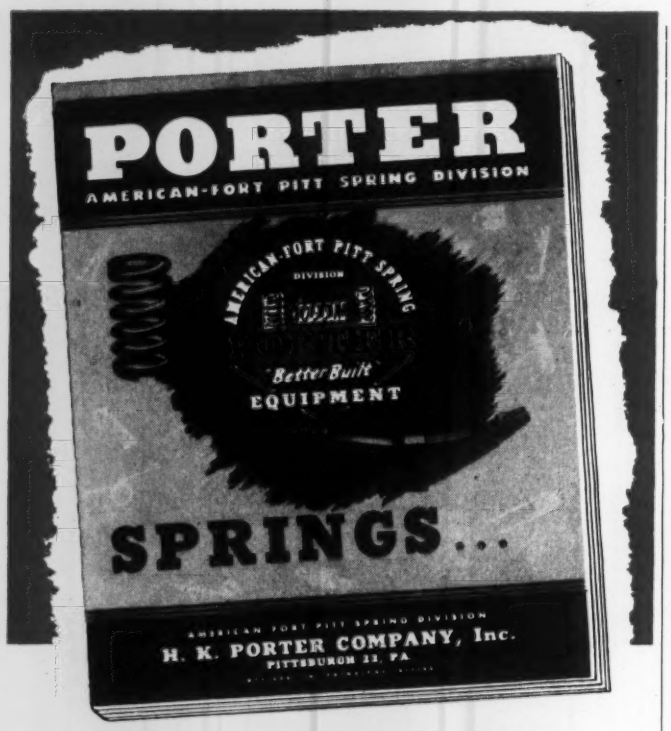

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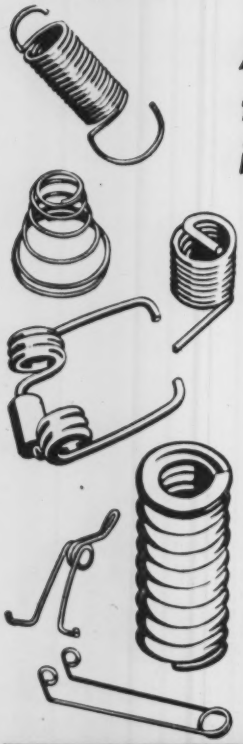

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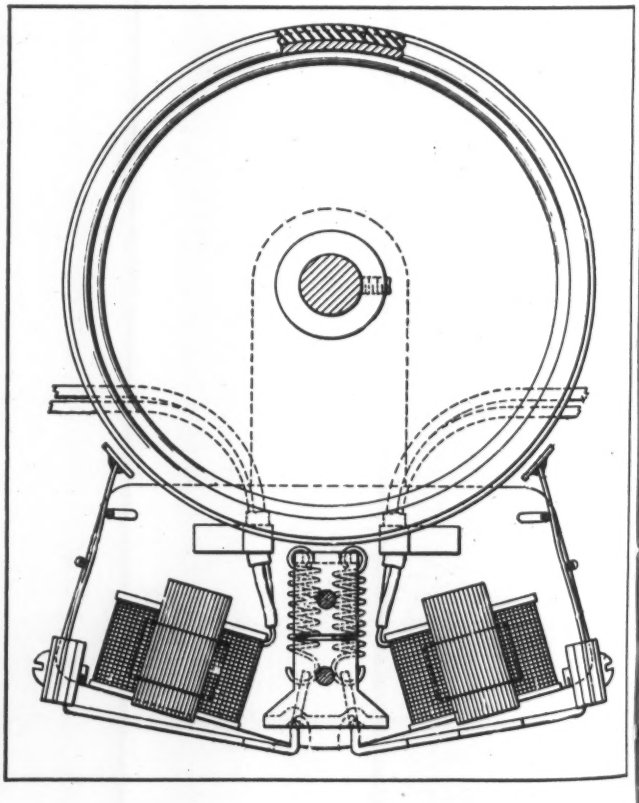
SMOOTH, RAPID ACCELERATION to desired speed without overrun is claimed for a unique positive-drive clutch covered in patent 2,427,292 assigned to Radio Corp. of America by Richard T. Mathes. Especially suitable for drives such as those for receiving distributors in multiplex telegraphy, the clutch arrangement locks in at the desired synchronous speed without overrun. Drive is accomplished by means of a conventional friction clutch, but overspeeding is obviated by a built-in toothed or overrunning clutch unit having the positive drive action opposite that usually employed.

FRICTION-DRIVEN electric motor which is exceptionally quiet in operation and which develops a relatively high torque and power output is covered in Alexander W. Plensler's patent 2,427,875. Assigned to Belmont Radio Corp., the novel motor comprises a rotatable driven wheel having a resilient peripheral friction surface and an electromagnet with a driving arm having a formed shoe attached to the extreme



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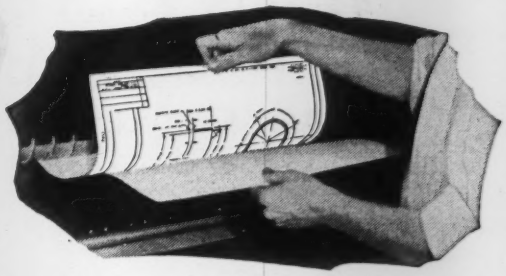

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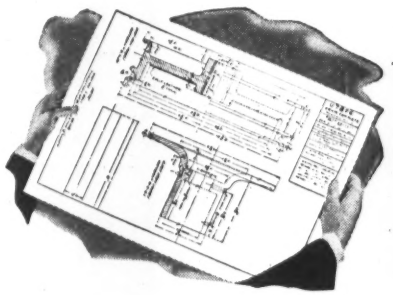

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STRAIN RELIEF BUSHING
to anchor the cord set to your electrically operated
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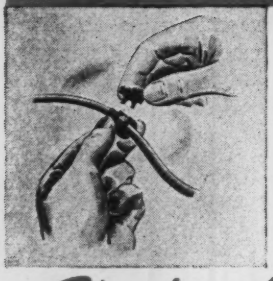
● The Nylon Heyco is an insulating grommet that is snapped into a hole in the chassis of a product. It anchors the cord set firmly to the chassis and imparts a positive non-slip grip—it does not injure the wire. Product life is greatly increased by preventing all strain on terminal connections as well as preventing cord wear at chassis entrance.



IT'S EASY TO APPLY HEYCO
1. Slip HEYCO on wire.
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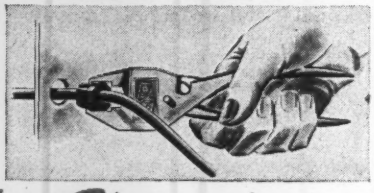

Here's what Heyco does-

1. Absorbs cord pull, push and torque
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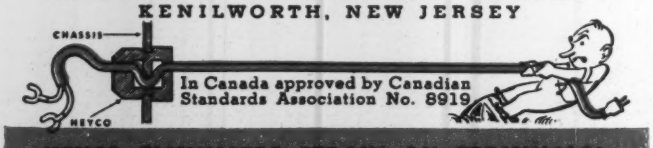

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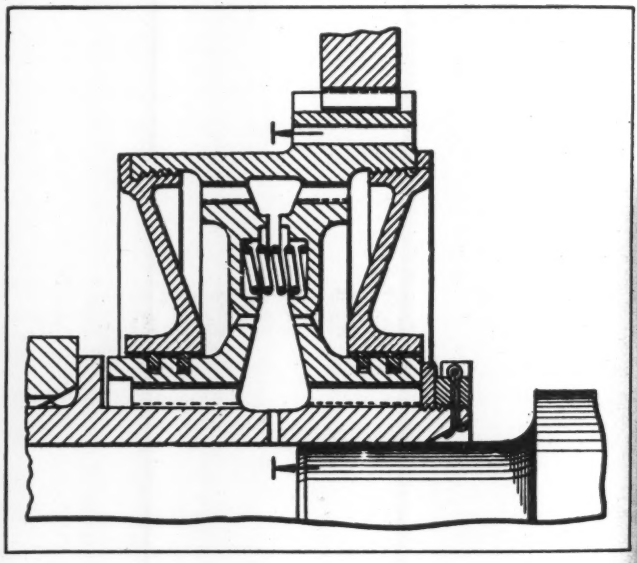
end for engagement with the wheel surface. Energization of the electromagnet with 60-cycle alternating current produces a low-amplitude, 120-cycle oscillation of the driving shoe producing continuous rotation of the wheel. Two driving units make possible reversible operation.

VIBRATIONS created in transmission of power from an internal-combustion engine such as in aircraft, etc., are largely eliminated by an unusual damper assigned to United Aircraft Corp. by Philip P. Newcomb. Covered under patent 2,427,505, the damper is designed to be entirely housed within a single drive unit and is light in weight and compact. A pair of pistons splined to the drive shaft and surrounded by oil are coupled by means of helical splines



on their outer periphery to the driving gear case. Torsional vibrations in the drive shaft cause the pistons to move toward or away from each other due to thrust imposed by the helical splines on the pistons. Vibrations are absorbed by this pumping action of the pistons which forces the damping fluid in and out of orifices in the sides of the pistons as well as through the spline teeth.

RECIPROCATORY MOTOR, having a polarized armature which pulsates at a frequency corresponding to that of the energizing current, provides the power for an unusual hydraulic pump unit. Covered in patent 2,430,758 assigned to Crise Electric Mfg. Co., by G. W. Crise, the unit is primarily intended for use as a hydraulic lifter for governing the operation of controls such as those on furnaces. Mounted between springs, the armature pulsates up and down at normal current frequency to actuate the attached pump piston. When desired, the reciprocating motor can be designed to pulsate at one-half normal current frequency.

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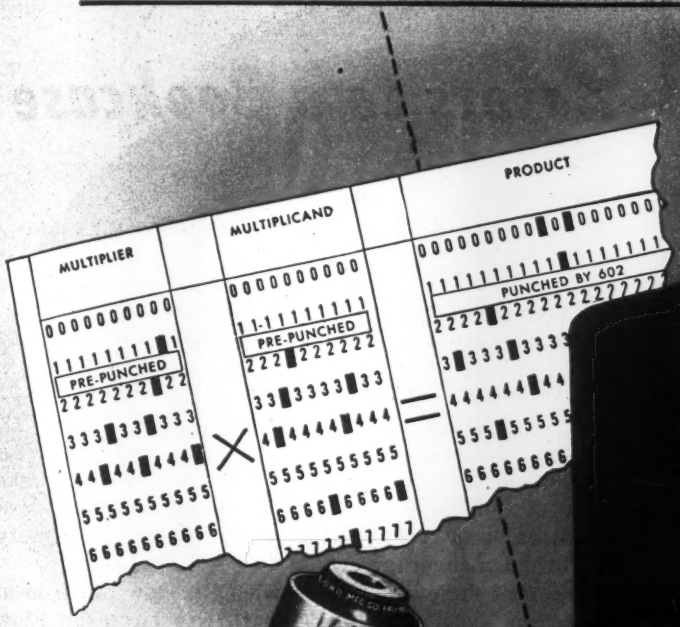

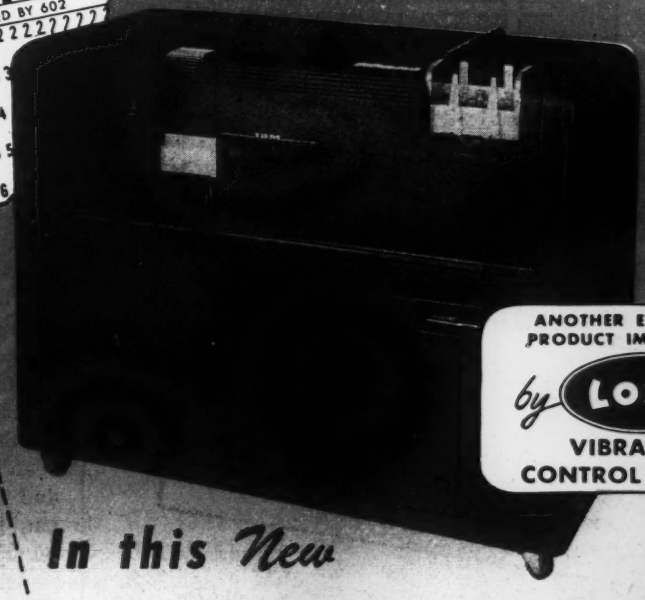

ANOTHER EXAMPLE OF PRODUCT IMPROVEMENT by LORD VIBRATION CONTROL SYSTEMS

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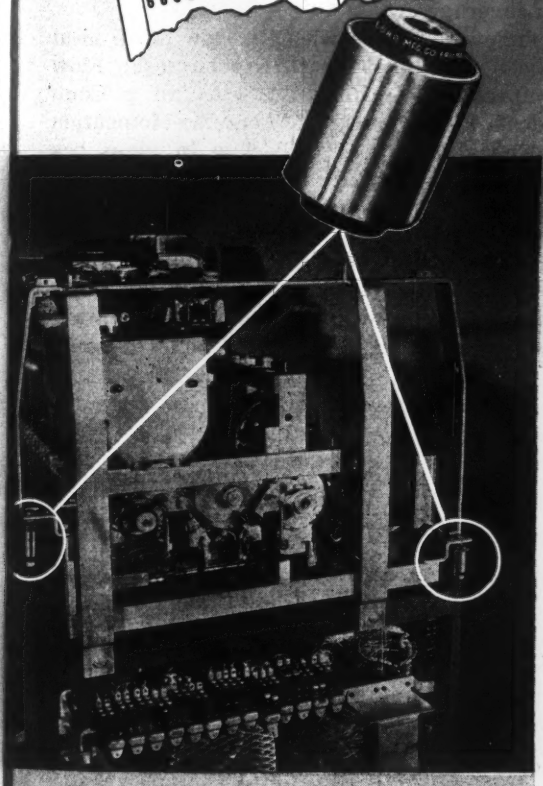

A marvel of inventive ingenuity that multiplies two ten-figure numbers and gives you the twenty-figure product in less time than it takes to tell it; that adds, subtracts, and divides with the same ease—that's the new IBM Calculating Punch.

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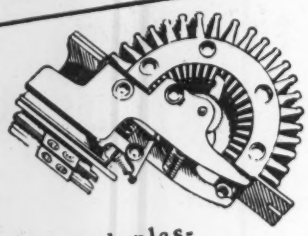

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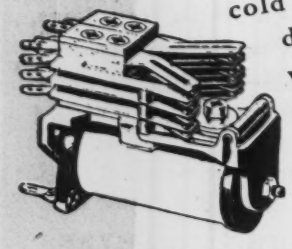

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Elementary Fluid Mechanics

By John K. Vennard, associate professor of fluid mechanics, Stanford University; published by John Wiley & Sons, Inc., New York; 339 *pages,* 5½ *by* 8¼ *inches, clothbound; available through* MACHINE DESIGN, $4.00 *postpaid.*

Published originally in 1940, this book has received considerable acclaim as a basic work in the fields of fluid statics and dynamics. It has now been rewritten and reorganized to increase readability and broaden coverage.

The first chapter, dealing with the flow of an ideal fluid, has been expanded to three chapters: Flow of an Incompressible Ideal Fluid, Flow of a Compressible Ideal Fluid and The Impulse-Momentum Principle. The chapter on ideal flow in pipes has been almost entirely rewritten to improve clarity, and the chapter on flow around immersed objects has been brought up to date with special emphasis on frictional drag.

An additional change in this second edition is the condensation of mathematical developments and the deletion of topics which the author believes to be adequately treated in other fields. The result is a shorter book which covers more material than the first edition.

□ □ □

Injection Molding of Plastics

By Islyn Thomas, president, Thomas Mfg. Corp.; published by Reinhold Publishing Corp., New York; 534 *pages,* 6 *by* 9 *inches, clothbound; available through* MACHINE DESIGN, $10.00 *postpaid.*

Inasmuch as this book is the first to be published on injection molding of plastics exclusively, the author has made it comprehensive in its coverage, treating production, die design, materials, and history, as well as design of parts. Nevertheless, the engineer involved to any degree in the design of plastic parts to be made by this process will find the volume an extremely valuable source of information.

The book is at once a text and a reference work in its factual sequence and its wealth of data. It discusses injection-molding techniques, materials, mold and product design, finishing and decorating of molded parts, and cost of production. In addition, it offers an entire section on physical properties, giving such values as: Tensile and compressive strength, modulus of elasticity, and specific gravity

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Silicone News

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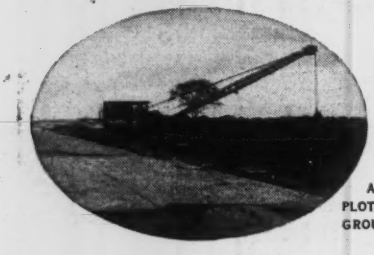
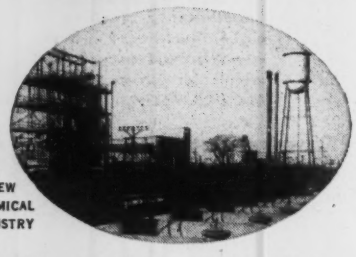
Now we're five years old. We can look back with some pride on what we've accomplished in that time. We look with equal satisfaction on the work done by our customers. Without their initiative no new material could have been adapted to so many different uses in so short a time. Working together, we have built a new chemical business to serve all industry.



1943

A PLOT OF GROUND



1948

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1948—We are now producing 60 different Dow Corning Silicone Products in the form of fluids and oils, greases and compounds, resins and varnishes and Silastic* all of which will do many things no other materials can do.
1943—We had essentially one customer and his name was Uncle Sam.
1948—Among our 7500 industrial customers are concerns in almost every major industry. Recently, we entered the consumers' market to bring silicones home to John Q. Public.
1943—We started to build a plant and a main office in Midland, Michigan.
1948—We have a large plant, now almost twice its original size, working full time to supply you with the silicone products you have come to depend upon. It's the first plant of its kind ever built. To pace our increasing productive capacity we now have branch offices in four major cities and two new ones about to be opened. In **1948** as in **1943**, we look forward, with your help, to a future busy with new silicone products and to a plant several times the size of our present one.

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for most of the important commercial plastics. Extremely well presented, the volume is replete with illustrations, drawings and tabulations.

□ □ □

Valve Gear Design

By Michael C. Turkish, cam-design engineer, the Wilcox-Rich Div. of Easton Mfg. Co.; published by that company in Detroit; 130 *pages,* 6¾ *by* 10 *inches, clothbound; available through* MACHINE DESIGN, $6.00 *postpaid.*

Possibly the first comprehensive discussion of internal-combustion engine valve gear, this volume covers the theory and design of cams, tappets, springs and other units associated with the dynamics of valving. Based on practice and experience as well as theory, the book is certainly an outstanding contribution to the literature.

Commencing with an analysis of the function of valve gear, the author proceeds through the subjects of cam design, contact stress, spring design and component study. The preponderance of the book, however, is devoted to cam design and its problems. Treated are such aspects as contours using flat or roller followers. Such lift curves as "constant acceleration," "harmonic," "contracted harmonic," and "secant" are discussed in detail. Numerous tables and charts greatly facilitate calculations while equations are carefully propounded and explained.

□ □ □

Materials Handbook

By George S. Brady, former chief materials consultant, Office of Civilian Supply; published by McGraw-Hill Book Co., New York; 831 *pages,* 5½ *by* 8½ *inches, clothbound; available through* MACHINE DESIGN, $7.00 *postpaid.*

First postwar edition of the well-known materials reference book, this volume will bring the reader up to date on developments in the period since 1944, presenting fresh, important facts on approximately 2000 materials which are either new or modified. This, the sixth edition, has been completely revised and contains discussions of over 7500 materials, together with important information of a more general engineering nature. An important change is the enlargement of what was the appendix in former editions to form part two of this volume. This second part occupies fifty pages and is devoted to the economics of materials and tabulations of basic measurements such as metal gages, hardnesses and temperatures.

Tool Steel Simplified by Palmer and Luerssen, is a discussion of alloy steel, the quality of which is best exemplified by the extreme popularity which it has enjoyed in the past. Such was the demand for the

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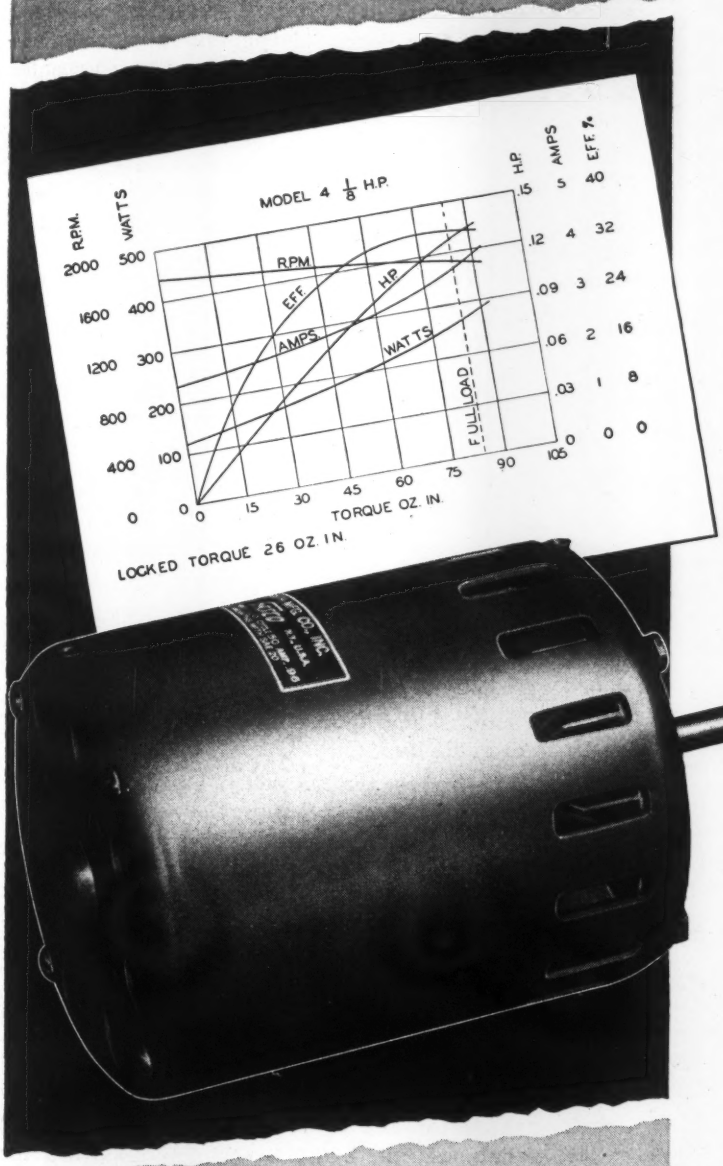

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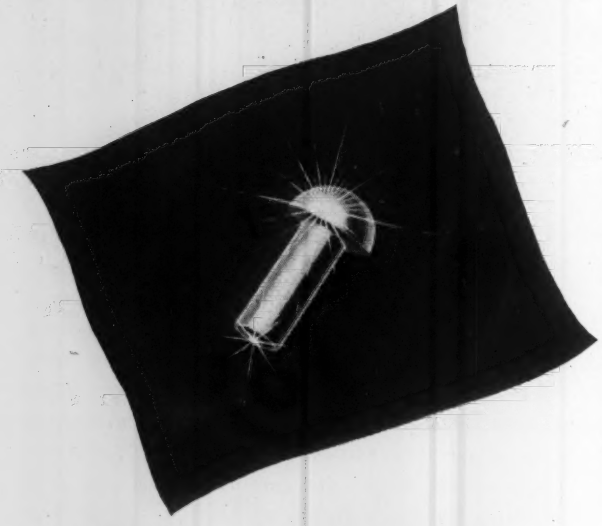

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first edition, published in 1937, that fourteen additional printings were made in the following eight years. This, the second edition, has been greatly revised to bring it up to date and includes the newer alloys as well as modern theory and practice. For example, the two new steels in the Carpenter Red-Hard matched set and the new Air-Hardening matched set are discussed in detail. The newer concepts of hardenability are treated at some length in chapter 4, while such subjects as heat-treating methods and hardenability tests are discussed in their respective chapters. Three new chapters have been added to the book. Chapter 9 presents the "tool steel selecter", chapter 19 covers the heating time of tool steels and chapter 13 is devoted to high-speed and hot-work steels. The 6 by 9-inch clothbound book has 564 pages; it is available at $2.00 from The Carpenter Steel Co., Reading, Pa.

CORRECTION: The book "The Chemistry and Technology of Plastics" by Raymond Nauth, reviewed in the January issue, Page 176, was erroneously described as "The Chemistry and Technology of Resins."

Shell Type Motors

(*Continued from Page* 162)

temperature of the coils when cold resistance is measured.

As has been mentioned, the machinery manufacturer is entitled to co-operation from the motor manufacturer. Approval and criticism of the proposed method of ventilation by motor engineers often saves headaches because the motor engineer has made airflow tests, heat runs, and ventilation studies for the standard motors and should have much data and experience to guide him.

BALANCE: Today, the majority of complete motors are dynamically balanced. Vibration amplitude limits are set for standard motors, and the machinery manufacturer can obtain particularly precise balance for an increased price. Shell-type rotors are dynamically balanced on balancing arbors, but regardless of how well a rotor is balanced prior to mounting on the final shaft, unbalance may result from the fit, shaft or bearing eccentricity, load unbalance, critical speed of the system, or from other causes. In normal rotor construction, rotors are pressed or shrunk onto the shaft before machining. This definitely anchors the rotor and when it is finally balanced, it stays in balance. In many built-in applications the rotor is held onto the spindle with a nut and lockwasher because the rotor must be removable. This method of mounting may introduce stresses and unbalances unless the clamping collar and rotor faceplate are a precise fit. Many shafts are sprung when the nut is tightened excessively. Final balance of the complete assembly is a necessity in precision tools for good operation. In doing this, material should not be removed from the rotor and rings.

In the past, critical speeds have not been much of

First All-Welded Crankcase Reduces Costs 54% ... Improves Appearance

By M. V. Caldwell, President
Globe Mfg. and Compressor Co.
Battle Creek, Michigan

THE redesign of our air compressor crankcases was born of necessity. Past difficulties in obtaining parts of the conventional construction for the original design prompted us to convert to welded design and led to what we believe is the first all-welded crankcase used in the compressor industry. The changeover was accomplished gradually, one step at a time so as not to affect production schedules.

Fig. 1 shows the original construction of cylinder and crankcase assembly for our 20 C.F.M. air compressor. This unit had a total weight of 151½ lbs. and cost $29.57 for material and direct labor. In the first step toward welded construction, the lower crankcase only (Fig. 2) was welded and assembled to the cylinder and upper crankcase castings. The total weight of this design is 85 lbs., and its cost is $25.81 for material and labor.

Our success with the welded lower crankcase encouraged a further development shown in Figs. 3 and 4. This all-welded crankcase weighs 23 lbs. and costs only $13.76. A comparison of the original cost of $29.57 with this reduced cost of $13.76 represents an over-all production saving of 54%.

The component parts for the all-welded crankcase are fabricated from ⅛" and 3/16" sheared mild steel plate as shown in Fig. 3. The parts are clamped solidly in a universal jig for downhand welding, and then we weld inside and outside with 3/16" "Fleetweld 5" electrode using 300-amp. Lincoln "Shield-Arc" Welders.

Several indirect benefits are gained with our welded design. In order to reach the required production, new and more elaborate patterns for grey iron would have been required at a cost of approximately $4,000 besides the expensive heavy fixtures necessary in machining the unwieldy cylinder casting. This cost has been avoided. In addition, an exceptionally high scrap loss, originally experienced in the lower crankcase, has been eliminated by the welded steel construction.

Savings made in weight are important to us because our selling prices are made f.o.b. the destination. Also reduction in base size from 20" x 26" to 16" x 22" is a great advantage in export shipping costs. Last, but by no means least, the welded construction simplifies painting and finishing to produce a more pleasing appearance and smoother finish.

Fig. 1. Cylinder and crankcase assembly originally used. Cost $29.57.

Fig. 2. Intermediate step in welded development. Welded lower crankcase assembled to upper crankcase and cylinders.

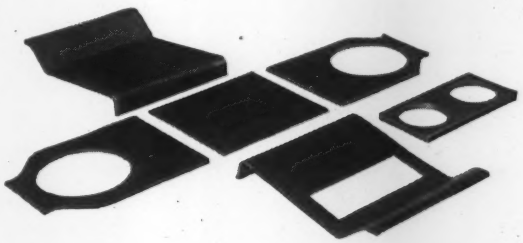

Fig. 3. Fabricated steel parts for all-welded crankcase.

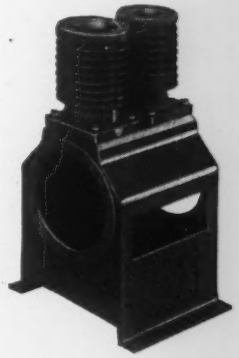

Fig. 4. Present all-welded crankcase and cylinder assembly. Cost $13.76.

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RESISTORS · RHEOSTATS · RELAYS · CONTROL DEVICES

a problem on small motors except in unusual cases where the motor had to power a badly unbalanced load. With very high speeds, such as 60,000 or 120,000 rpm, critical speeds must be watched carefully. For precision grinding, it is much better to operate below the main critical speed than above it. Therefore, these motors must be made "stiff", that is, with large shafts, small rotor diameters, and short bearing spans. In addition, the center of the load should be as close to one of the bearings as possible. At the same time, bearings should be as small as possible so that the high bearing speed will not result in greatly reduced bearing life. Sometimes the inner race of the bearing can be machined on the shaft to keep shafts larger and bearing speeds down.

BEARINGS AND LUBRICATION: Because ball bearings are generally used with shell-type motors, it is suggested that the machine designer confer with a reputable bearing manufacturer to obtain recommendations covering bearing types, mounting and lubrication. Regarding lubrication, care should be taken to see that oil or grease cannot get out of the bearings and onto the motor windings with resulting decreased insulation life.

Special Requirements for High Speeds

Frequently at high speeds, special bearings with plastic or bronze retainers are used, grease lubricated. If speeds are extremely high, oil lubrication of the bearings becomes necessary, with a predetermined rate of feed. Frequently oil-mist lubrication, supplied by a pressure vapor system, is employed at extremely high speeds and special, super-precision bearings are used to obtain the required accuracy. However, for this type of application the recommendations of a bearing expert should be obtained.

MOUNTINGS: Uniform air gap results in uniform torque, good characteristics, and a quiet motor. An uneven gap exerts a side pull on the shaft, which adds to the deflection of the shaft and gives rise to noise and vibration. Therefore, concentricity to close tolerances must be maintained, with the air gap variation not exceeding 10 per cent of the gap. Shell diameters are ground to a tolerance of plus 0.000, minus 0.002-inch; shaft bores are ground to tolerances of plus 0.0000, minus 0.0005-inch, and the planes of endplates are machined perpendicular to the shaft axis within 0.001 to 0.002-inch at the largest diameter of the facing, depending on the size of the motor. Concentricities of the bore with the shell diameter and of the rotor bore with rotor diameter are held to close tolerances by the manufacturer of the motor.

Machinery manufacturers should keep shaft runouts, stator housing bores and rabbets, and end bracket bores and rabbets to equally close limits to insure proper motor operation. Shafts must be designed with enough strength to prevent deflection in excess of 0.001-inch. Critical speeds are lowered inversely proportional to the square of the deflection. Provision should be made so that the motor can be easily dismantled in case of emergency, and also so that the bearings can be properly lubricated. Motor connections also should be readily accessible.

MEETINGS AND EXPOSITIONS

Apr. 19-21—

American Society of Lubrication Engineers. Third annual convention and exhibit to be held at the Hotel Statler, Buffalo. W. F. Leonard, 343 South Dearborn St., Chicago 4, is secretary-treasurer.

Apr. 26-27—

Association of Iron and Steel Engineers. Annual spring conference to be held at Buffalo. Additional information may be obtained from society headquarters, 1010 Empire Bldg., Pittsburgh 22. T. J. Ess is managing director.

Apr. 26-30—

American Management Association. Seventeenth packaging exposition and conference to be held at the Cleveland Public Auditorium, Cleveland. Clapp and Poliak Inc., 350 Fifth Ave., New York 1, is exposition management.

May 3-7—

American Foundrymen's Association. Fifty-second annual convention and exhibit to be held in the Philadelphia Convention Hall, Philadelphia. William W. Maloney, 222 West Adams St., Chicago 6, is secretary-treasurer.

May 20-21—

Society of the Plastics Industry. Annual meeting to be held at Atlantic City, N. J. Additional information may be obtained from the headquarters of the society, 295 Madison Ave., New York 17. William T. Cruse is executive vice president.

May 20-22—

American Society of Mechanical Engineers. Oil and Gas Power division meeting to be held in St. Louis, Mo. Additional information may be obtained from society headquarters, 29 West 39th St., New York 18. C. E. Davies is secretary.

May 27-29—

Society for Experimental Stress Analysis. Spring meeting to be held at the Roosevelt Hotel, Pittsburgh. W. M. Murray, P. O. Box 168, Cambridge 39, Mass., is secretary-treasurer.

May 30-June 5—

American Society of Mechanical Engineers. Semiannual meeting to be held at Milwaukee. Additional information may be obtained from the headquarters of the society, 29 West 39th St., New York 18. C. E. Davies is secretary.

May 31-June 2—

American Society of Refrigerating Engineers. Thirty-fifth spring meeting to be held at the New Ocean House, Swampscott, Mass. David L. Fiske, 40 West 40th St., New York 18, is secretary.

June 6-11—

Society of Automotive Engineers. Summer semiannual meeting at French Lick Springs Hotel, French Lick, Ind. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

June 7-9—

American Gear Manufacturers Association. Thirty-second annual meeting to be held at the Homestead, Hot Springs, Va. Newbold C. Goin, Empire Bldg., Pittsburgh 22, is executive secretary.

June 14-18—

American Society for Engineering Education. Fifty-sixth annual meeting to be held at the University of Texas, Austin, Texas. Additional information may be obtained from Prof. Arthur B. Bronwell, secretary of the society, Northwestern Technological Institute, Evanston, Ill.

June 17-19—

American Society of Mechanical Engineers. Applied Mechanics division meeting to be held at Chicago. Additional information may be obtained from society headquarters, 29 West 39th St., New York 18. C. E. Davies is secretary.

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Ward Leonard 130 Relay easily adapted to a wide variety of requirements

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-TO GEAR BUYERS

FREE THIS HANDY GEAR TOOTH CALCULATOR

The numbers on the outer edge of this device represent the diametral pitch of the gear. By turning the cut-out section to the desired number, the figures appearing in the space show circular pitch, circular thickness, addendum and the whole depth. • These calculators are available to gear buyers upon request—with our compliments.

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PERKINS Precision, Custom-Cut GEARS

PERKINS MACHINE & GEAR CO., Springfield 2, Mass.

BUSINESS AND SALES BRIEFS

HUNTER PRESSED STEEL CO., Lansdale, Pennsylvania has inaugurated a series of free training courses on spring testing and quality control. Requests by management men for enrollment in the course are invited. Starting April 6, the courses will be two days in length and cover parts-inspection techniques and simplified statistical methods of analysis of data.

♦

V & E Mfg. Co. of Pasadena, California has recently appointed Roland D. Tyler sales manager. Mr. Tyler is responsible for assisting Vemco distributors in their sales development programs.

♦

Redmond Co. Inc. has appointed Ralph Redmond director of sales. Mr. Redmond is also vice-president and treasurer of the corporation. Prior posts held by him include those of purchasing agent and director of purchases.

♦

American Pulley Company's board of directors has elected Frank E. Brown to the office of secretary and treasurer; he succeeds Gonzalo C. Munoz who has retired. Caleb F. Fox III, vice-president, will relinquish responsibility for production to succeed Mr. Brown in charge of sales and advertising.

♦

Simmons Fastener Corp., has announced that Harry H. Rose has been named general sales manager. Prior to this appointment, Mr. Rose served as sales engineer.

♦

Chain Belt Co. has made the following changes in its executive staff. L. B. McKnight and O. W. Carpenter have been elected directors of the company, Mr. McKnight was also elected a vice-president of the company with responsibility for the two heavy machinery divisions of the company. Mr. Carpenter was also elected a vice president of the company in charge of finance. In addition, B. F. Devine was elected vice

t.f.e. it's the only way to cut fastening costs

You pay more money to specify, purchase, stock, inspect, requisition and *use* fasteners than you do to buy them. True Fastener Economy means making sure that *every* function involved in the *use* of bolts, nuts, screws, rivets and other fasteners contributes to the desired fastening result—maximum holding power at the lowest possible *total* cost for fastening.

You Get True Fastener Economy When You Cut Costs These Ways

1. Reduce assembly time with accurate, uniform fasteners
2. Make satisfied workers by making assembly work easier
3. Save receiving inspection through supplier's quality control
4. Design assemblies for fewer, stronger fasteners
5. Purchase maximum holding power per dollar of initial cost
6. Lower inventory by standardizing types and sizes of fasteners
7. Simplify purchasing by using one supplier's complete line
8. Improve your product with a quality fastener.

True Fastener Economy is the Lowest Total of All Fastening Costs.

103 Years Making Strong
the Things That Make America Strong

RUSSELL, BURDSALL & WARD
BOLT AND NUT COMPANY

Plants at: Port Chester, N. Y., Coraopolis, Pa., Rock Falls, Ill., Los Angeles, Calif. Additional sales offices at: Philadelphia, Detroit, Chicago, Chattanooga, Portland, Seattle. Distributors from coast to coast.

Simplifies Design Problem

"The Development Superintendent was amazed by the way I simplified the problem of designing our new textile machine," says P. W. "By using stock products listed in the Boston Gear Catalog, I cut design time 'way down. There were 101 groups of products and 4326 separate stock items listed — with specs and data on each. It certainly cut out a lot of engineering work . . . and we really beat our budget figures on that job!"

Saves Drafting Time

"A design job came to my board marked 'RUSH' . . . with the due date only 3 days away," writes F.R. "It seemed impossible because the driving elements looked plenty complicated. But in checking my Boston Gear Catalog, I found the whole drive could be made up of stock parts. All I did was copy off the specs from pages 28, 126, 168 and 254. That job got done in six hours!"

Send in your experience in cost, time, or work savings made possible by the Boston Gear Catalog.

Manufacturer Cuts Costs

E. P. L. writes: "I give the Boston Gear Catalog direct credit for boosting my profits. For years we'd made every component in our mixing machines. But one day, looking for a quick source of high-quality commercial gears, I checked the Boston Gear Catalog and found drive components that exactly filled my needs . . . and at a lower price than we could make them! When I found I could get quick delivery from a nearby distributor, I standardized on the Boston Gear line. The savings in engineering and production time, plus lower unit costs, add up to higher profits for me."

Put this great data book to work earning for You!

FREE This valuable data book which shows you how to make important savings in design, manufacturing and maintenance costs using Boston *Stock* Component Machine Parts. Contains 101 separate groups of products — gears, chain, sprockets, speed reducers, etc. — 4326 individual items — engineered, mass produced to custom standards for your application. Also contains much valuable working data for designers, engineers, maintenance men.

Name ..... Position .....

Company .....

Street .....

City ..... Zone ..... State .....

BOSTON GEAR WORKS

One of the world's most complete lines
62 distributors in major cities

18 Hayward Street
Quincy 71, Massachusetts

president; he will continue his duties as manager of the construction-machinery division.

♦

C. P. Clare & Co. has recently moved its New York office. It is now located in suite 310 of the Graybar building, 420 Lexington Ave.

♦

Jenkins Bros. has appointed Robert V. Salisbury manager of its New York branch. Mr. Salisbury, formerly assistant to the general sales manager in New York, will have his headquarters at 80 White St.

♦

Ohio Nut & Bolt Co. has acquired patents covering the brazed assembly of metal parts. Facilities for utilizing this process will be housed in new quarters which will be built at an early date.

♦

Willys-Overland Motors, drop forge division has appointed J. R. Lakin general sales manager. Mr. Lakin will co-ordinate sales, advertising, sales promotion and public relations.

♦

Saval Co. has announced the promotion of J. E. Flickinger to the position of sales manager. Prior to this time, Mr. Flickinger had been in charge of the company's aircraft sales in the western territory.

♦

Dayton Rogers Mfg. Co. has recently appointed Marvin C. Barnum as their New York representative.

♦

The Cleveland Graphite Bronze Co. has organized its facilities for the manufacture of heavy-duty types of engine bearings into a single separate unit. Manager of the new unit will be M. J. Merlin. The change will facilitate manufacture of this type of heavy bearings and enable the company to meet present heavy demand.

♦

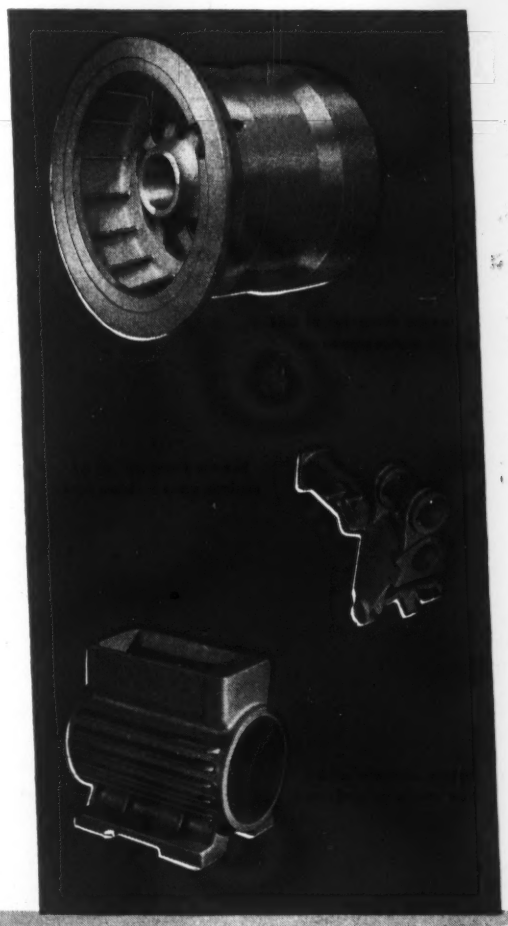
Shakeproof Inc. has appointed Charles F. Keyser Jr. manager of distributor sales. This position has been created to expand merchandising and sales assistance to distributors of the fasteners, and to facilitate co-operative sales effort between distributors and Shakeproof salesmen. Mr. Keyser's office will be at 2501 N. Keeler Ave., Chicago 39.

♦

Union Mfg. Co. has acquired the air chuck and air chuck cylinder business of the Hannifin Corp. Accessories

Typical Example of WELLMAN LEADERSHIP in Light Metal Casting:

AVIATION



These wheels, landing gear parts, and engine parts, produced for major aircraft companies, are typical of the type of work we do in magnesium and aluminum.

Any application requiring magnesium, aluminum or bronze castings can be handled from blue print stage to production . . . whether it's aircraft, domestic appliances, pneumatic tools, machinery castings or any of the other countless places where light metals should be used.

Our quotations on your casting requirements are backed by 36 years' experience; complete laboratory control from raw metal to finished castings; three plants; X-Ray inspection; and unlimited know-how.

Just send us your blueprints — or ask us to send our field representative.

"Wellman means Well-Cast."

Patterns in Wood & Metal

THE **WELLMAN** BRONZE & ALUMINUM COMPANY

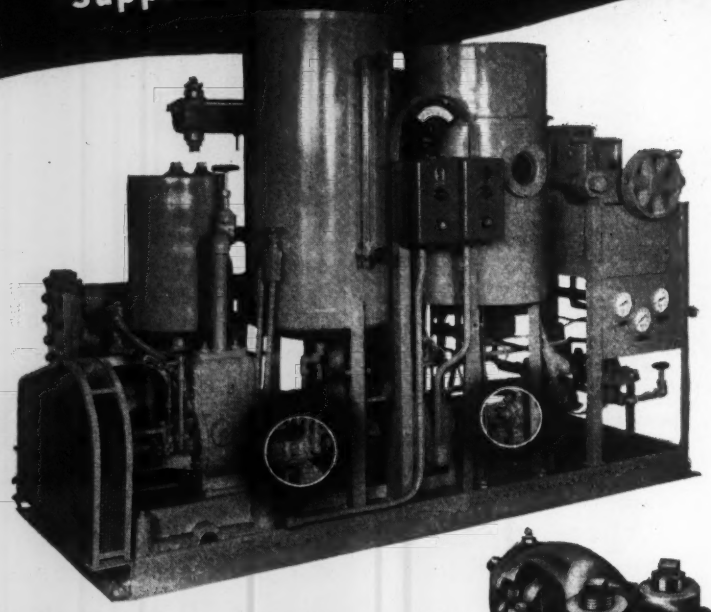
GENERAL OFFICES:

2547 EAST 93rd STREET • CLEVELAND 3, OHIO

CASTINGS X-RAY INSPECTED

SUCTION

For Buckeye Oil-Purifying Process Supplied by Roper Pumps



Roper Pump Meets Challenge of 28+" Vacuum

Buckeye Laboratories Corporation of Cleveland, Ohio, makers of equipment for cleaning, dehydrating, degasifying and stabilizing oil, install Roper Pumps as integral parts of their equipment. On the machine illustrated above, a Roper Pump transfers oil under pressure to the atomizing nozzles which in turn spray the oil mist into the vacuum chamber.

As the processing is completed another Roper pumps the purified oil from the vacuum chamber. This requires exceptional pumping efficiency to maintain a flow against the resistance of a 28 + " vacuum.

Another example of Roper adaptability to an almost limitless range of uses.

Roper 4-port design offering 8 different piping connections cuts installation time and costs to a minimum . . . 4 piping arrangements with pump operating clockwise and 4 other arrangements for counter-clockwise operation.

Ropers run quietly and efficiently in either direction. They are designed along the correct hydraulic principle and built compactly to fit a wide range of application and mounting conditions.

Send for Bulletin

Illustrating and describing pumps built to handle pressures up to 1000 lbs. p.s.i., capacities ¾ to 300 g.p.m. at speeds up to 1800 r.p.m.

GEO. D. ROPER CORPORATION

244 Blackhawk Park Ave.
ROCKFORD, ILLINOIS

ROPER
Rotary Pumps

as well as chucking fixtures will be furnished by this New Britain, Conn. firm.

♦

Badger Plastics, Inc. has recently sold its extrusion division to the Defiance Machine Works Inc. The injection division of the company has been sold to the Plastics Engineering Co. of Sheboygan, Wis.

♦

U. S. Electric Motors, Inc.'s Atlantic plant at Milford, Conn. is headed by Frank M. Mason, newly appointed vice president. Mr. Mason will make his headquarters at Milford.

♦

Burke Electric Co. has announced discovery in active service of the first unit made by the company, a motor-generator set manufactured in 1907. F. E. Mahew, chief electrician of the Crystal City plant of the Pittsburgh Plate Glass Co., who had this unit under his supervision, was rewarded for his discovery by being given the "Oldest Burke" award.

♦

E. I. du Pont de Nemours & Co.: Newly elected chairman of the board of this company, Walter S. Carpenter Jr., has resigned as President. He is succeeded by Crawford H. Greenewalt former vice president and vice president of the executive committee.

♦

Gray Iron Founder's Society, northern Ohio group, recently elected Henry J. Trenkamp chairman. At the same time James B. Heisler was elected vice-chairman and F. J. Dost was named secretary-treasurer. Mr. Trenkamp is president of the Ohio Foundry Co., Mr. Heisler is vice president of A. C. Williams Co. and Mr. Dost is vice president of Sterling Foundry Co.

♦

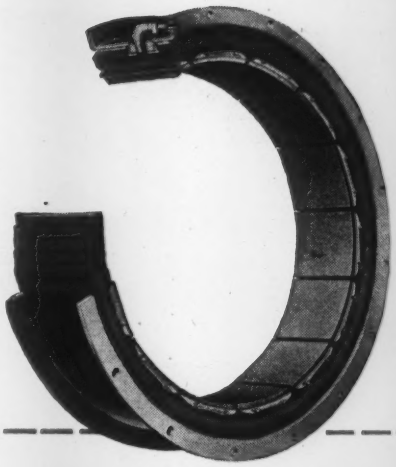
Western Gear Works president Thomas J. Bannan was recently elected a national vice president of the National Association of Manufacturers. Mr. Bannan served as vice chairman of the resolutions committee of the NAM and has recently completed his term of office as president of the American Gear Manufacturers association.

♦

John Ek Industries Inc., Guilford, Conn., manufacturers of gasoline engines and automatic equipment has elected Dr. Haig Solakian president;

Low Maintenance Cost . . .

High Machine Production



The Dollars-and-cents Advantages of the Airflex Principle

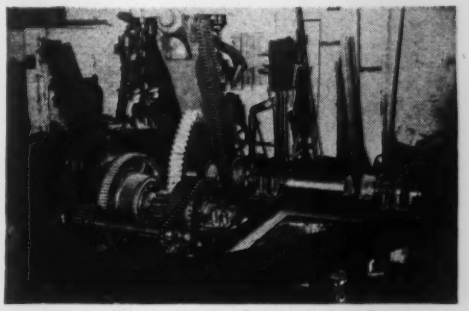
The experience of thousands of satisfied users proves that Fawick-equipped machines do more work—at less cost—and earn more money.

Fawick Clutches adjust themselves. They put an end to wasteful clutch-adjustment down time. They contain no arms, levers or springs to wear or break. In fact, they reduce clutch maintenance costs practically to zero.

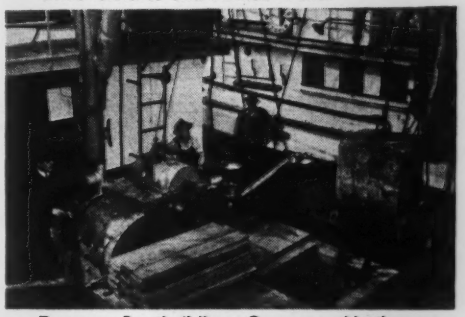
Furthermore, a Fawick Clutch is a shock absorber that protects the entire machine, reduces over-all maintenance expense and adds years to its productive life. It cushions and absorbs shocks, jars, jolts and vibration, preventing unnecessary breakage, undue wear and tear and needless parts replacement costs.

The Airflex principle has proved its advantages under exacting conditions in many fields—petroleum, earth-moving, metalworking, rubber, pulp, paper and others. New applications are being developed constantly.

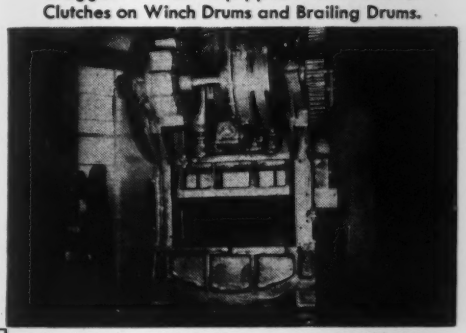
Write our Engineering Department for recommendations of the Fawick elements best suited for your equipment.



Draper Engine Works Company Log Turner equipped with Fawick Forward and Reverse Drive Clutches and Fawick Brake.



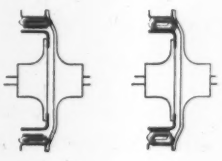
Tacoma Boatbuilding Company Northern Dragger Winch equipped with Fawick Clutches on Winch Drums and Brailing Drums.



Rockford Iron Works Company Model 6½-60 Press equipped with Fawick Clutch and Brake.

HERE'S HOW IT WORKS

Compressed air expands the rubber-and-fabric tube to engage clutch with any degree of "grip" you want. Release the air and clutch disengages.



FAWICK *Airflex* and AIRING **CLUTCH**

FAWICK AIRFLEX CO., INC. 9919 Clinton Rd., Cleveland 11, O.

In Canada, Renold-Coventry Ltd., Montreal, Toronto, Vancouver, Quebec • In Britain, Crofts Engineers, Ltd., Bradford, England

CASTINGS BY *Continental*

THIS JOB MAY SOLVE A *Casting Problem* FOR YOU!!!

There may be immediate help for one of your design problems in the foundry "know-how" of CONTINENTAL. The job shown above may be one you'll never tackle, yet it illustrates one of the thousands of end-products in which the skilled precision and economical dependability of CONTINENTAL Castings are important factors. Almost no matter what you design, CONTINENTAL'S three complete foundries and machine shops can provide unmachined or finished carbon or alloy steel castings that will save time and money in your end-product manufacture or material costs. For problem-solving counsel on castings — submit your early plans for CONTINENTAL recommendations and estimates.

Design with an eye to supply!

Special heavy duty Fast's coupling for 30" dia. shaft. Weighing 22 tons, this coupling is designed for transmitting 10,000 h.p. at 17 RPM. Manufactured by the Fast's Coupling Division of Koppers Co., Inc., Baltimore . . . Castings by Continental!

Continental
FOUNDRY & MACHINE CO.
CHICAGO • PITTSBURGH
Plants at: E. Chicago, Ind.; Wheeling, W. Va.; Pittsburgh, Pa.

Mr. Solakian resigned his office of vice president of the A. F. Holden Co. to accept this presidency.

♦

Allis-Chalmers Mfg. Co. has recently announced two changes in its sales representation. George H. Carden has been named manager of the Baltimore district office and T. G. Smith has been appointed resident representative at Beaumont, Texas.

♦

Allen-Bradley Company's line of motor controls will be sold in northwestern states by the R. A. Lundbeck Co. of 1900 University Ave., St. Paul, Minn. Area covered includes Minnesota, North Dakota, South Dakota, and the western counties of Wisconsin.

♦

National Formetal Co. has recently elected Anne Jacobsen vice president. Mrs. Jacobsen has been with the organization five years, starting as a private secretary.

♦

Thomas Machine Manufacturing Co. has appointed Paul E. Lunquist sales engineer and special field representative. Prior to joining this organization, Mr. Lunquist was a design engineer for Chicago Bridge and Iron Co.

♦

Electro Metallurgical Co. has made several new appointments in its technical service departments. Among the new division managers of service are: J. N. Ludwig Jr. for the Pittsburgh area, R. J. Portman for Chicago, F. W. Hanson for New York, and W. B. McFerrin for Detroit.

♦

Whitney - Hanson Industries Inc. has recently been formed by merger of the Whitney Chain & Mfg. Co. and The Hanson-Whitney Machine Co. Winthrop H. Whitney is chairman of the board of the new company. Each organization will operate as a separate division of the corporation.

♦

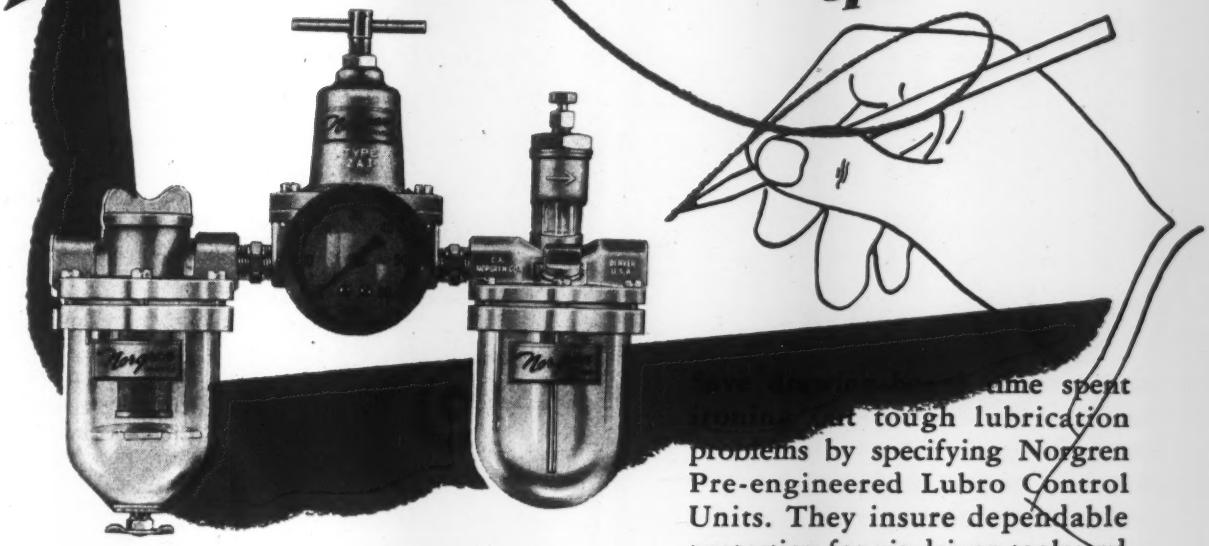
John S. Barnes Corp. of 144 Walnut St., Rockford Ill., has announced appointment of a new sales representative in the east. Handling sales in the middle Atlantic states and including New Jersey, Delaware, Maryland and the eastern half of Pennsylvania as well as metropoitan New York will be H. E. Stone Supply Co., 2 E. Haddon Ave., Oaklyn, N. J.

♦

Powdered Metal Products Corp., Franklin Park, Ill., has announced the appointment of E. Richard Wal-

TAILOR MADE. . .

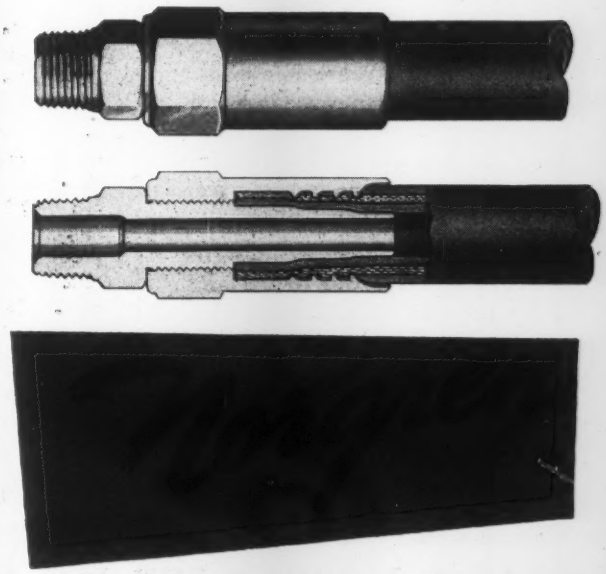
to meet your drawing board specifications



Save [illegible] time spent [illegible] tough lubrication problems by specifying Norgren Pre-engineered Lubro Control Units. They insure dependable protection for air driven tools and cylinders. Unit consists of filter, regulator and lubricator; *cleans, controls* and *lubricates* air for power packed results. Add this to your performance factors for maximum salability. Send your specifications for complete information.

REUSABLE . . . *yet failure-proof far beyond the burst point of the hose!*

That's the remarkable, new reusable coupling engineered by Norgren for hydraulic assemblies. No more expensive inventories of stand-by hose assemblies. Bulk hose and a few Norgren reusable couplings are all you need to make efficient, quick replacements of hydraulic assemblies.



CHECK THESE SUPERIOR FEATURES

- **LASTING REUSABILITY**
- **MINIMUM OF DOWN TIME**
- **SOLID BAR-STOCK BRASS**
- **ELIMINATES STAND-BY INVENTORIES**
- **FOR HOSE FROM ¼" TO 1" I. D.**

WRITE FOR DETAILED INFORMATION

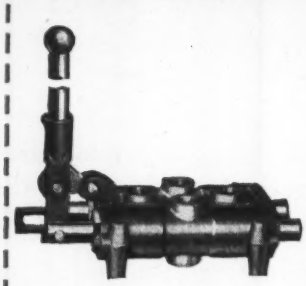
C. A. NORGREN CO., 222 SANTA FE DRIVE, DENVER 9, COLORADO

Service built in beneath the surface

● It isn't what you see—but the working parts you can't see that's important about control valves.

Positive and fast acting with U-shaped packers that are expanded by the pressure, Quick-As-Wink Control Valves seal tightly preventing leakage and bleeding. All operating parts are in pressure balance eliminating any tendency to creep or crawl. There is no lapping—no grinding—no metal to metal seating. Every valve is a high quality control—precision made—and individually tested before being shipped. This assures users a maximum of long, efficient, trouble-free service.

Furnished in hand, foot, pilot, solenoid and diaphragm operated designs for controlling all types of air and hydraulic equipment. Let us work with you on your requirements!



Quick-As-Wink Lever Operated Hydraulic Valves

The MF-651-N4 pictured above is a four-way neutral position valve, for controlling double acting cylinders. It can also be furnished with three position "compound-exhaust", or "compound-on" actions. Recommended for use with oil or water up to 150° F. and 2000 PSI. Other types for pressures up to 5000 PSI, — but send for a catalog today and get *full* details about the *complete* line.

Quick-As-Wink Control Valves

Manufactured by **C. B. HUNT & SON, Inc.,** *Salem, Ohio.*

Engineering and Sales Representatives in the Principal Cities

ter as sales manager. Mr. Walter was formerly chief metallurgist of the organization.

♦

Ampco Metal, Inc. has made a number of changes in its sales organization. In the Detroit area, J. K. Bybee has been appointed district manager with Don Coleman assisting as field engineer. The Industrial Instrument Co. of Millbury, Ohio will serve as Toledo area representative, while Gordon E. Brown is the new district manager for the Chicago territory where he is assisted by Robert H. Louthain.

♦

American Brakeblok Div. of the American Brake Shoe Co. has elected William T. Kelly Jr. its first vice president. In adition to his new duties, Mr. Kelly will continue as president of the Kellogg and the Engineered Castings divisions.

♦

Penn Electric Switch Co. has appointed Howard C. Shilling district manager of the Goshen, Indiana territory. Formerly, Mr. Shilling was manager of the company's Moline office.

♦

Federal Machine and Welder Co. of Warren, Ohio, has announced the election of a number of new officers. Allen R. Kelso is the newly elected executive vice president and general manager of the Warco Press and Warren Stamping divisions; Walter Dinse is vice president and general manager of the Welder division, and Henry A. Stix is treasurer and comptroller.

♦

Engineering Products Co. of Los Angeles has appointed a new manager of sales, Howard Ledeen. Mr. Ledeen will be in charge of the sales of Ledeen heavy-duty cylinders.

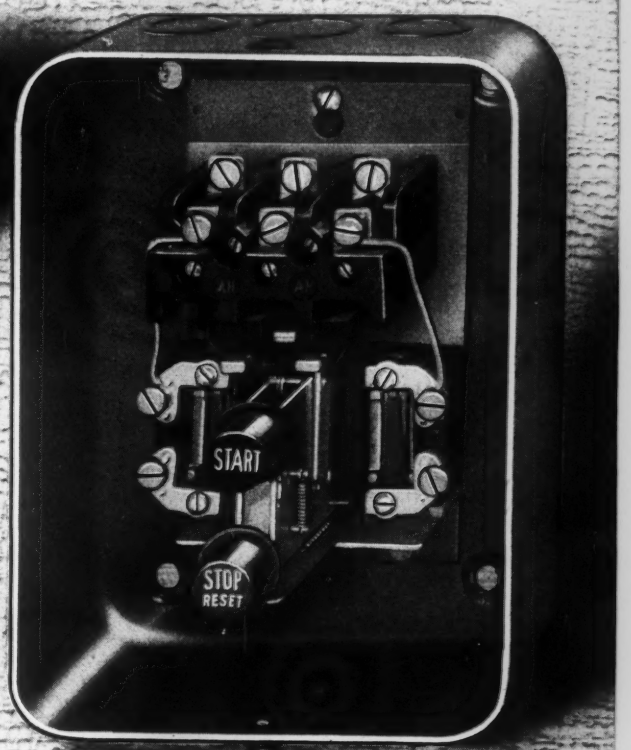
♦

Flexible Tubing Corp., organized in 1947, has announced its entry into the flexible tubing field with design engineering and manufacturing facilities in Branford, Conn. President of the new corporation is Frederick K. Daggett, former sales and engineering manager of the Spiratube division of the Warner Brothers Co.

♦

The Bellows Co. has announced realignment of sales territory into five regional sales divisions. They will be known as eastern, midwestern, central, southern and western. First of the regional sales manager appointees is Harold P. Granger, who will head the eastern division with offices in Boston. Midwest division will be under the supervision of J. James Mudd, formerly assistant sales manager of the company.

MANUAL RESETTABLE STARTERS



Type "RT"
SIZES 0 and 1

ACROSS-THE-LINE TYPE, push-button operated for motors up to 7½ H. P., 600 Volts maximum, A. C. The individual unit construction enables parts replacements to be made from the front. All line and load connections made also from the front. Switch is readily removed from the enclosure.

Thermal overload unit — with interchangeable heaters — is trip-free of the mechanism, handily accessible from the front. Easily-replaced contacts, silver-to-silver double break; butt contacts self-adjusting and self-aligning. Quick make-and-break.

Available in 2, 3 or 4 poles, Type "RT" Starter proves to be the *starting-point* of trouble-free motor operation. Write us about those requirements of yours for which this switch seems particularly well designed.

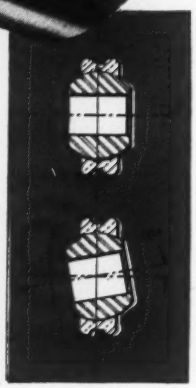
ARROW AH HART

THE ARROW-HART & HEGEMAN ELECTRIC COMPANY, HARTFORD 1, CONN., U.S.A.

BEARINGS

Maximum 10 degree permissible mis-alignment ...ability to withstand ultimate static bearing loads to 700,000 lbs...shock resisting characteristics. These features make the Halfco bearing the answer to your design problems. The Halfco bearing consists of a heat-treated, hard chrome-plated, highly polished steel ball around which an outer race of hard bronze is integrally formed. This full spherical surface contact allows extremely heavy loading and resists both axial and radial forces. The self-aligning and axial thrust capacities make the Halfco bearing far superior to plain journal bearings. A 360 degree oil groove assures positive lubrication. Various sizes permit a wide variety of design possibilities.

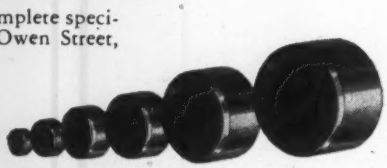
Halfco rod end bearings are ideal for mechanical linkage systems. They may be used as ball joints, rod ends on actuating cylinders, or in linkages to accommodate motion in more than one plane.



BRIEF SPECIFICATIONS

Ultimate static load capacities, lbs.	3,500 to 700,000
Bore sizes	.1895" to 3.000"
Permissible mis-alignment	approx. 10°

Write for catalog containing complete specifications Address 10738 Van Owen Street, Burbank, California.



HALFCO BEARINGS are manufactured with either aluminum bronze or steel races in a wide variety of sizes.

ADEL

ADEL PRECISION PRODUCTS CORP.

BURBANK, CALIF. ★ HUNTINGTON, W. VA.

Manufacturers of: Aircraft Hydraulic Systems • Marine & Industrial ISOdraulic Controls • Halfco Self-Aligning Bearings • Line Support Clips and Blocks • Industrial Hydraulic Equipment • Aircraft Valves • Industrial Valves

NEW MACHINES

And the Companies Behind Them

Agricultural

DUSTER. For mounting on any tractor; steel construction. Two compartments in dust hopper for different insecticides. Nozzles adjustably spaced on boom. Large ball-bearing fan driven from side pulley on tractor. The Stacey Co. Inc., Dallas 1, Texas.

Communication

TRANSCRIPTION PHONO-P.A. UNIT. Portable. For 16-in. transcriptions and commercial recordings. Dual-speed turntable; crystal pickup; two microphone inputs; heavy duty 8-in. permanent-magnet speaker. Power consumption: 54 watts. For 117 volts, 60 cycles. Bell Sound Systems, Inc., Columbus, O.

Domestic

HEARING TESTER. Portable. For individual or group testing. Spring-wound motor maintains constant record speed; battery-powered vacuum tube amplifier; lightweight crystal pickup. Western Electric Co., New York 7.

SLIDE PROJECTOR. Handles film and 2 x 2-in. slides. Film fed by neoprene rollers; slide film holder completely detachable; easily replaced 150-watt lamp. Radio Corp. of America, RCA Victor Div., Camden, N. J.

Heat Treating

GAS-FIRED FURNACES. Protective atmospheres such as Exogas, Endogas, Monogas, and Ammogas. For annealing, carburizing, drawing, etc. Cylindrical bell types, continuous pusher types, continuous roller hearth types, and continuous conveyor types. Westinghouse Electric Corp., Pittsburgh 30.

Manufacturing

PROJECTION WELDER. Delivers 58,000 amps with power demand of 348 kva at 61 per cent power factor. Box type frame design; rocking or stationary lower electrode holder; upper holder raised and lowered by air cylinder. Taylor-Winfield Corp., Warren, O.

AUTOMATIC WOOD-GLUING PRESS. Generator for dielectric heating; I-beam construction, shielded by aluminum sheet. Controls, coils, and wiring protected by filter capacitors. Pressure on platens by hydraulic cylinders. Weldon-Flinchbaugh Co., York, Pa.

BRAKE SHOE WELDER. Joins brake

"For the Blower...
I'M INSISTING ON THE
RAYTHEON 480
MOTOR"

RAYTHEON MOTORS FOR SPECIAL APPLICATIONS supplied in H. P. ratings from 1/30 to 1/8. Single, also variable speeds from 1,100, 1,700 or 3,400 R.P.M.; Outside diameter—5"

When it comes to fan or blower applications—more and more engineers are specifying the Raytheon Model 480 Motor. Here's why: it can be supplied for open or totally enclosed installations—with a single or double shaft extension. Versatile? You bet! Can be furnished as a single value capacitor—a two value capacitor—or as a split phase type.

Balanced mechanical design—plus high efficiency and remarkably low noise level are features that add up to outstanding performance under the most challenging conditions. Tell us about your motor problem. We'll be glad to help you solve it!

RUSSELL ELECTRIC COMPANY

344 W. Huron Street • Chicago, Ill.

SWIFT • SURE • STURDY • SILENT • SMOOTH • STEADY

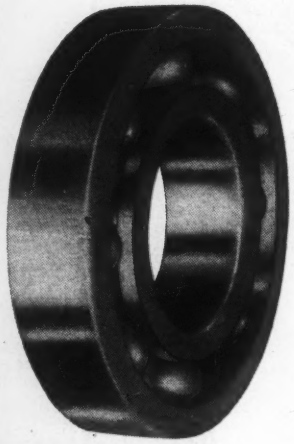
PLAN Swift FALCON-FAST ACCELERATION IN EVERY BASIC DESIGN

You've often remarked that no piece of equipment can be superior to its component parts...

You appreciate the broad recognition and acceptance of BCA Ball Bearings in industry...

You know you can tell your engineers to draw upon our half-century of experience in friction control practices, encompassing many varied fields of industry...

You, who insist upon continuous, full-capacity, trouble-free operation, know this can be assured by specifying BCA Ball Bearings.



BEARINGS COMPANY OF AMERICA
LANCASTER, PENNSYLVANIA

BCA RADIAL · ANGULAR CONTACT · THRUST BALL BEARINGS

shoes to rims. Power supplied by transformer, 300 kva at 50 per cent duty cycle. Roller bearing head; weld pressure applied hydraulically; safety switch prevents operation without work in position. Thomson Electric Welder Co., Lynn, Mass.

INCLINED WELDING PRESS. Multiple spot welding type. Open-back, four-post construction; air-operated ram; individual spring-backed electrodes; anvil type lower die. Thomson Electric Welder Co., Lynn, Mass.

BAND SAW. Fourteen-inch. Fully enclosed; electric-welded steel framework. New patented blade guides; one control knob for setting side jaws and ball-bearing backup roller. Worktable 15 by 15½ inches. Boice-Crane Co., Toledo 6, O.

RADIAL SAW. Motor, one or two hp, operates ac-de, 25 to 60 cycles, with spindle speed of 8000 rpm. Twelve-inch diam. saw blade; length of stroke, 27 in. Worktable, 17 by 46 in.; weight, 260 lb. Porter-Cable Machine Co., Syracuse, N. Y.

HAND GRINDER, SANDER AND WIRE BRUSHER. Vertical type; 6000 to 8000 rpm free speed. Magnesium housings; positive governor control and on-off throttle control. One-piece spindle in ball bearings; cast-in muffler; automatic lubrication. Master Pneumatic Tool Co., Inc., Orwell, O.

TOOL ROOM MICROSCOPE. Revolving plate head. Table travel, 4 in.; table tilt, ±10 deg; measurements to 0.00005-in.; magnification power, 30 x; objective field, 5/16-in.; net weight of unit, 120 lb. E. Leitz Inc., New York 13.

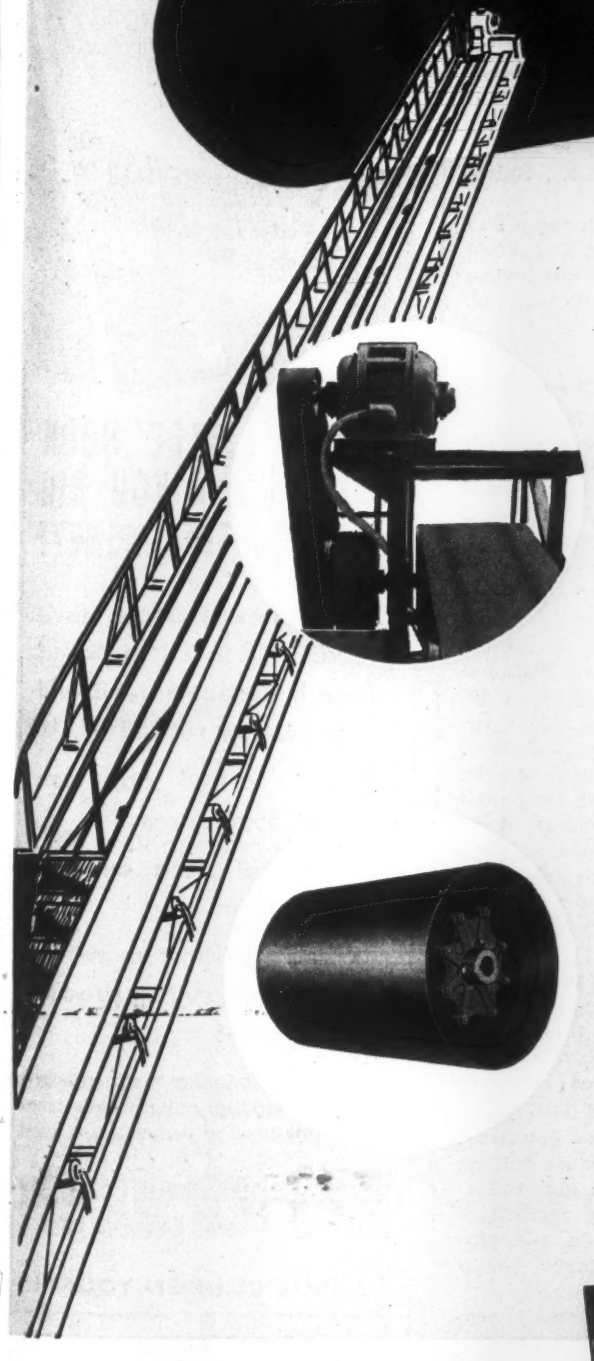
OPTICAL COMPARATORS. For comparing magnified shadows of parts with master outlines. Eight models, having tables from 12 to 16 inches, and magnifications from 5 to 125. Jones and Lamson Machine Co., Springfield, Vt.

ABRASIVE WHEEL DRESSER. Operates on 10 to 1 ratio from template which controls movements of diamond through stylus. Vital parts dust-protected with neoprene boots. Capacity, 10-in. diam, 1⅛-in. wide wheel. Hoglund Engineering Co. Inc., Union, N. J.

AUTOMATIC ASSEMBLING MACHINE. Assembles packing washers onto grooves in hydraulic brake pistons; 50 assemblies per min. Continuously revolving, variable-speed table driven through spur and worm-gear transmission; 2 hp motor. Base, welded steel; floor space, 45 by 45 in. Snyder Tool and Engineering Co., Detroit.

SPOT AND PROJECTION WELDER. To 200 strokes per min. Transformer con-

HOW YOU CAN MODERNIZE AND SIMPLIFY CONVEYOR DESIGN



By combining *American* Reduction Drives and *American* Conveyor Pulleys you get many unique advantages that help you modernize appearance, improve performance, and reduce installation and operating costs of the conveyors you design.

American Reduction Drives save space—the Unit mounts directly on the shaft of the driven machine. These Drives are compact, weigh less than ordinary drives, are neater in appearance. No "old-fashioned" supports or foundations are needed . . . installation is usually made in a matter of minutes. Any driven speed below 154 rpm is quickly obtainable; and once on the job, *American* Reduction Drives reduce maintenance costs because the primary belt drive absorbs damaging shock loads.

Rugged *American* Welded Steel Conveyor Pulleys combine light weight with high strength for long service life. Their true-running qualities result in efficient performance, less belt wear. They are easily, quickly installed because of their interchangeable hubs—and when combined with *American* Reduction Drives, installation of the complete Unit can be made with assembly line speed. Best of all, *American* Reduction Drives and *American* Conveyor Pulleys are standard, stock items!

Decide now to add all these design and production benefits to the conveyors you design. It's easy to get complete selection and installation data—just drop us a line for descriptive literature.

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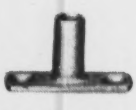

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★ TURN OUT FINER FINISHED PRODUCTS

UNITED-CARR FASTENER Corp.
CAMBRIDGE 42, MASSACHUSETTS

MAKERS OF DOT FASTENERS

nected to tap switches without flexing primary leads. Composite copper-steel knee construction. The Taylor-Winfield Corp., Warren, O.

Materials Handling

PORTABLE BELT CONVEYOR. Fifteen feet long; ends raised and lowered by handwheel. Belt, 24-in. wide, 3-ply solid woven cotton duck, is reversible. Sage Equipment Co., Buffalo 13.

FORK TRUCK PUSHER RACK. Powered by two hydraulic cylinders having equalized pressure. Forward travel of rack, 55½ inches, reduceable to meet special needs. Clark Equipment Co., Tructractor Div., Battle Creek, Mich.

PALLET TRUCK. Fitted with ramp and winch. Winch fitted with 15-ft steel cable; ramp hinged and removable. Lyon-Raymond Corp., Greene, N. Y.

Metalworking

BILLET SHEAR. 1200 ton. All gears and motor drive overhead; all moving parts enclosed. Cast steel housing; air-operated multiple-jaw clutch; cast steel cut herringbone gears run in oil bath. Thomas Machine Manufacturing Co., Pittsburgh 23.

GEAR SHAVING MACHINE. Employs principle of rotary crossed-axis shaving. Cutter located above work; fast-lock tailstock facilitates loading and unloading. National Broach & Machine Co., Detroit 13.

AUTOMATIC DISINTEGRATOR. For removing taps, drills, studs, reamers, etc. Revolving-head type; can be swiveled to any angle or compound angle. Extended-arm radius, 4 ft. Automatic feed and control; control panel equipped with voltmeter, control knobs and water valves. Ansaldi Tool & Engineering Co., Detroit 8.

GRINDING WHEEL SALVAGING MACHINE. Reworks grinding wheels to 42 in. diameter, 14 in. thick. Powered by 5-hp d-c motor. Weight, 10,000 lb. Wickes Bros., Div. of The Wickes Corp., Saginaw, Mich.

TUBE CUTTING AND FLARING MACHINE. For copper and aluminum tubing. Over 5000 flares in eight hours; also performs swedging operations. Tubing sizes, ¼ to ¾-in. OD. Motor, 1/3 hp. C. J. Unger Manufacturing Co., Dayton, O.

DIE CASTING MACHINE. Small production model. Metal injection by air pressure. Dies opened and closed by hand. Up to 500 cycles per hr. H. L. Harvill Mfg. Co., Corona, Calif.

ENGRAVING UNIT. Attaches to milling machine to permit two and three-dimensional profiling, contour milling and engraving. Reproduction ratios, 1:1 to 4:1. Ball bearings

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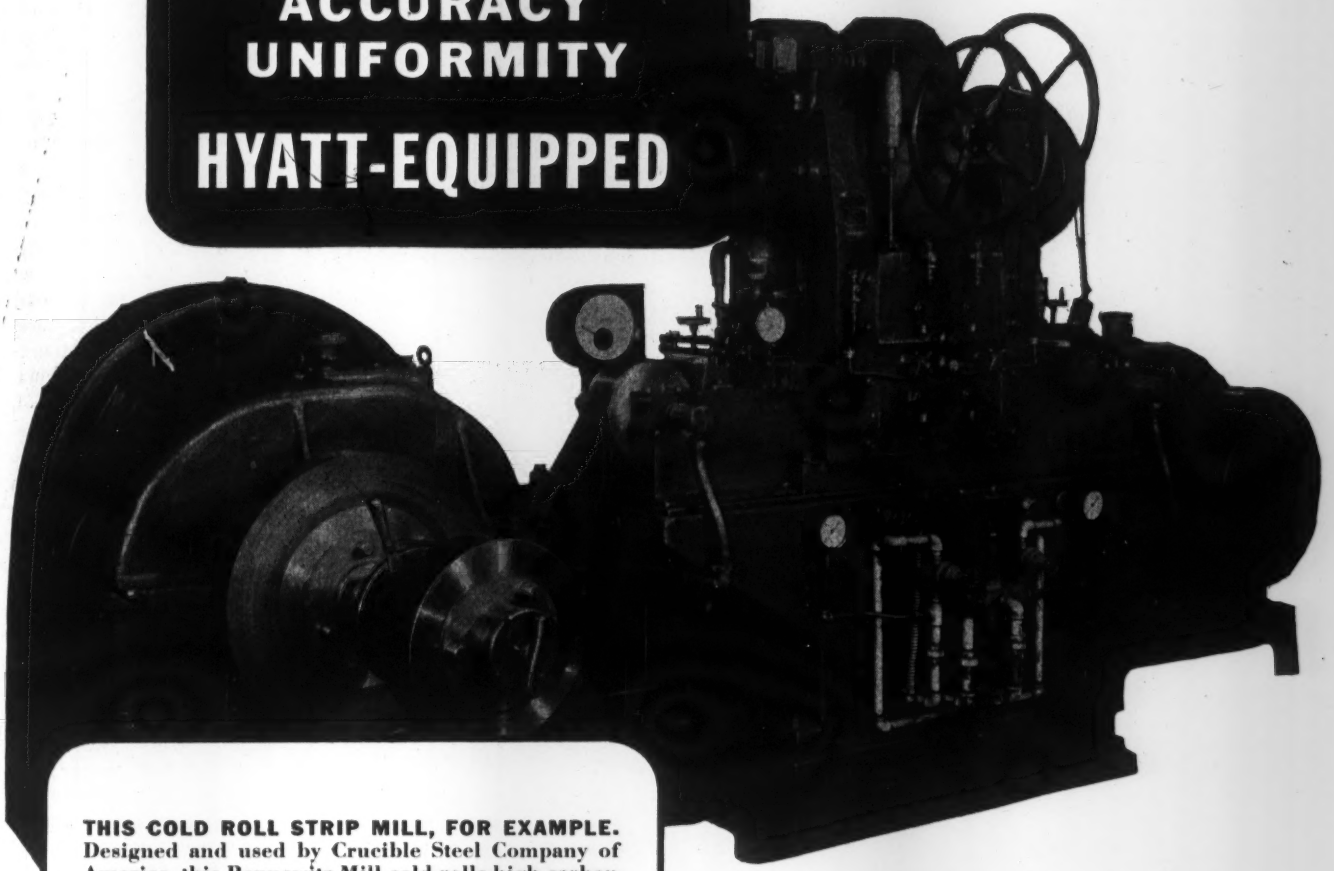
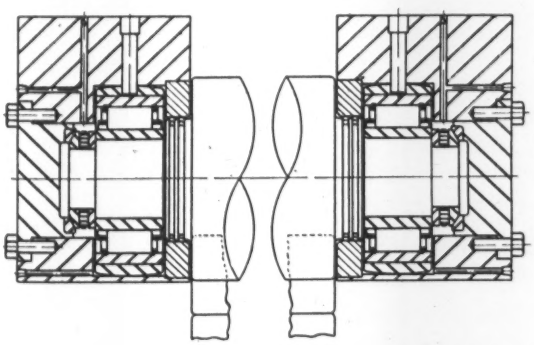


THIS COLD ROLL STRIP MILL, FOR EXAMPLE. Designed and used by Crucible Steel Company of America, this Bennewitz Mill cold rolls high carbon steel in coils 3½" wide, to thicknesses as low as .003" with tolerance on thin sizes held to plus or minus .0002". A precision operation, with remarkably high degree of finish and uniformity.



Two Hyatt Hy-Load Roller Bearings are applied to each of the four back up rolls, and provide rigid support for the work rolls. This is the result of the ingenious method of mounting . . . plus important features inherent to the Hyatt Roller Bearings themselves: their high capacity—uniformity of rollers and races—and minimum eccentricity.

From ore to finished product—in steel mills and in every branch of industry—you'll find thousands of successful applications, like this, with Hyatt Roller Bearings contributing to better design and better performance.

Fitting the bearing to the job—not the job to the bearing—is the Hyatt way of engineering. Thus, when Hyatt Roller Bearings are built into a machine, you have the assurance that the bearing is planned, manufactured and applied for your specific requirements. Hyatt Bearings Division, General Motors Corporation, Harrison, New Jersey.

HYATT ROLLER BEARINGS

Centrifugally-cast Ampco Metal valve-seat inserts are quality features of the 61 and 74 cu. in. O.H.V. engines on the famous Harley-Davidson motorcycles.
Durable
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–because it has the wear-resistance to "take it" on the toughest jobs

When a famous motorcycle manufacturer specifies Ampco Metal for valve-seat inserts *he knows his customers are getting a "plus" value.* And that value is featured by the manufacturer in advertising literature as an important sales appeal!

That's logical when you consider the long life and trouble-free service you gain, when critical parts are made of Ampco metal—with its outstanding resistance to corrosion, compression, impact, fatigue, and wear. It has excellent bearing qualities, too, plus unique efficiency at extreme temperatures.

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Field offices in principal cities

throughout. Auto Engraver Co., New York 19.

PRECISION BORING MACHINE. One-piece bed; sealed-lubrication boring heads; self-contained hydraulic units piped to a centralized manifold plate. Simplex Machine Tools Div., Stokerunit Corp., Milwaukee 14.

POWER BENDING MACHINE. For pipe, angle iron, bars, etc. Hydraulic, ram type; portable; self-contained. Three hp; weight, 1900 lb. Wallace Supplies Mfg. Co., Chicago 14.

MOTORIZED CRUSHING DEVICE. Self-truing; for crush truing grinding wheels. Self-contained; two-speeds; carbide tool for dressing the form of close-grained cast iron. Mounts on table of wet type surface grinder. The Sheffield Corp., Dayton 1, O.

CRANKSHAFT TURNING MACHINE. Carbide tools; automatic; produces 35 crankshafts per hr. Front and rear cam-operated angular cross slides; longitudinal turning slides in back-tool housing; hydraulic feed to all slides. Main motor, 40 hp; net weight, 27,000 lb. Wickes Bros. Div., The Wickes Corp., Saginaw, Mich.

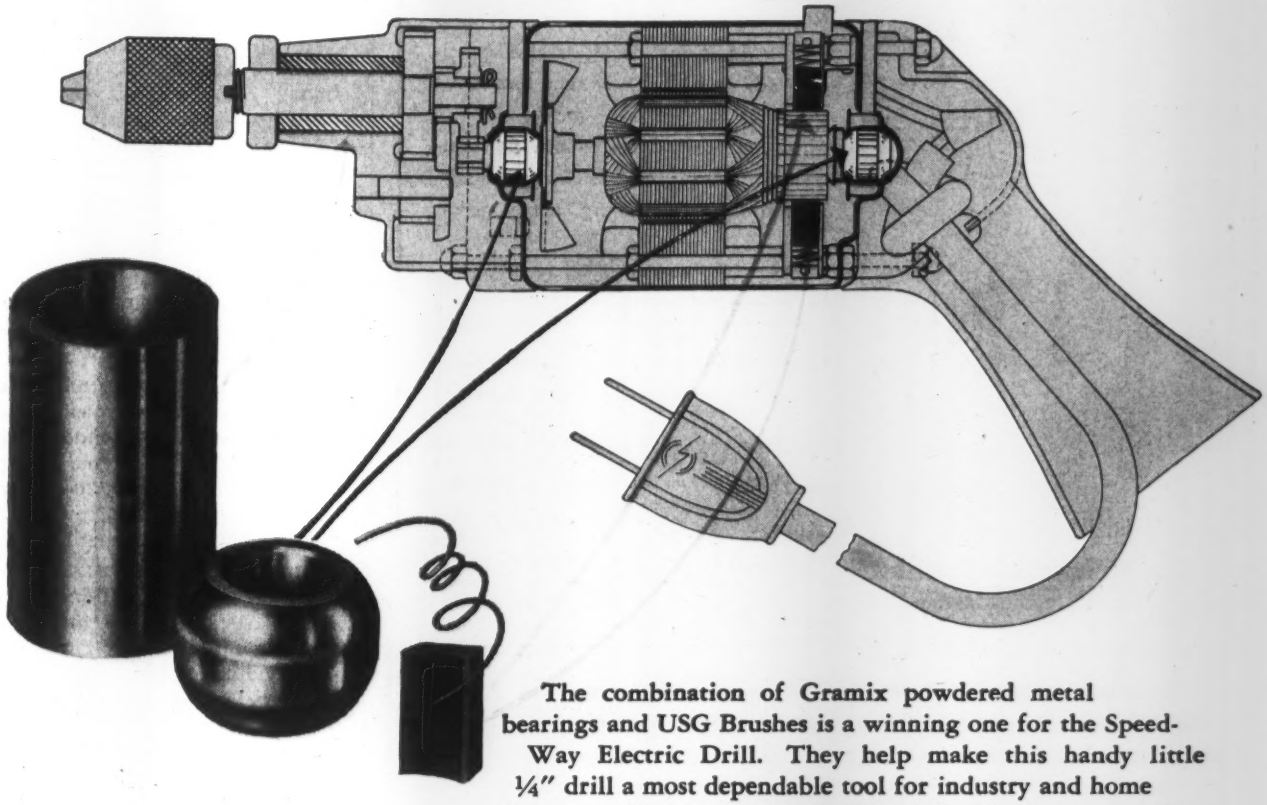
UNIVERSAL AND TOOL GRINDER. Three wheel-spindle speeds. Six table-speed rates, 7¾ to 100 in. per min; four headstock speeds, 106 to 525 rpm. Automatic lubrication of table ways and mechanisms in base by pump from oil reservoir. Separate motor drives for wheel spindle, table and headstock. Brown & Sharpe Mfg. Co., Providence 1, R. I.

VERTICAL MILLING MACHINE. Quick-change, full-geared table feed mechanism delivers power to table through splined feed shaft. Table, 8 by 34 in. Three table speeds, twelve cutter speeds to 2400 rpm. Spindle downfeed and table feed automatic. Right-angle milling head for horizontal milling including boring and drilling. Index Machine Co., Jackson, Mich.

SPECIAL MACHINE TOOL. Automatic. Drills, countersinks and taps heavy-duty truck rear-axle housings with one operator. Transfer mechanism carries work from operation to operation. Part automatically located and clamped at each station. Drills and countersinks hydraulically fed; taps lead-screw fed. The Cross Co., Detroit 7.

SPECIAL MILLING MACHINE. Thread-mills oil-well tool joints, continuous-cycle, four-station. Each station insulated from external vibration by unique spring-suspension mounting. Servo-controlled hoist lifts parts in and out of hydraul-

gramix bearings and USG brushes team up to provide top dependability for SpeedWay Drill



The combination of Gramix powdered metal bearings and USG Brushes is a winning one for the SpeedWay Electric Drill. They help make this handy little ¼" drill a most dependable tool for industry and home shop use. The strong, tough Gramix bearings — two self-aligning and one sleeve type — *keep shafts rotating free and true with a minimum of noise and wear.* Because they are porous, they retain lubricant and release it to the shaft as required. And because they are made by die-pressing powdered metals to exact size and with perfect bearing surfaces, they *do not require costly hand finishing or machining.* Hence, although they're more dependable, Gramix bearings cost less than conventional machined bearings. With USG Brushes also on the job, contributing their quiet, long-life performance, it's little wonder that this No. 69 drill is one of the most popular Blue Line tools produced by SpeedWay Manufacturing Co. of Cicero, Ill. Gramix bearings are not only used extensively in SpeedWay Blue Line Portable Electric Tools but also in its Silver Line Electric Motors.

We may be able to show you how Gramix powdered metal bearings, bushings, or other parts may *improve the performance of your product and save you money.* Send us prints for specific recommendations. Ask about our complete line of USG graphite, carbon-graphite, and metal-graphite brushes. **Write for your copy of the 264-page catalog of Gramix parts and specifications.**

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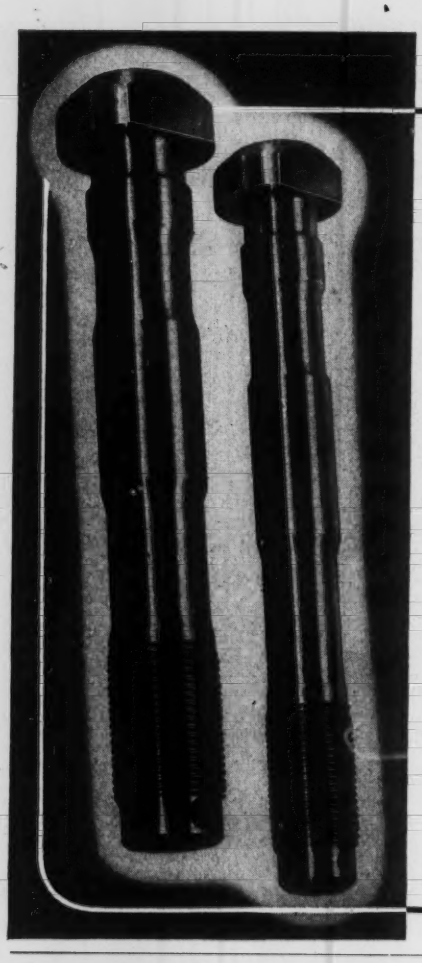

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MACHINE DESIGN

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ically-operated chucks. The Cross Co., Detroit 7.

ABRASIVE CUTOFF MACHINES. Dry type has 10-in. wheel on swinging arbor; 3-hp motor. Wet type has 16-in. wheel; 7½-hp motor. Tolerances to 0.003-in. Foot-pedal controlled work vises. The DoALL Co., Des Plaines, Ill.

DRILL PRESS. Precision; 14-in. Bench and floor models. Capacity ½-in. Built-in lighting; quick-acting belt-tension release lever changes spindle speeds. Spindle travel, 4 in.; spindle speeds, 707, 1305, 2345, and 4322 rpm; full-tilt table; motor, 1/3-hp, 1725 rpm. South Bend Lathe Works, South Bend 22, Ind.

VERTICAL MILLING MACHINE. Small, floor type; swivel or universal head; adjustable ram; positive spindle brake; spindle lock; screw-type spindle depth stop; 6 spindle speeds from 300 to 5000 rpm; motor, ½-hp, reversible. George Gorton Machine Co., Racine, Wis.

PLAIN MILLING MACHINE. Cutter can be automatically cycled to two different heights. Spindle-head feed rates, ½ to 7⅝-in. per min.; spindle-head fast travel, 30-in. per min.; brake-type motor. Brown & Sharpe Mfg. Co., Providence 1, R. I.

PLAIN GRINDING MACHINE. For cylindrical grinding; 10-in. swing; to 0.0001-in. tolerance in production; hydraulic and electrical controls; individual motor drives; automatic oiling. Brown & Sharpe Mfg. Co., Providence 1, R. I.

GRINDING MACHINE. To 16-in. diam swing; 36 to 168 in. between centers; work to 6500 lb. Hydraulic work traverse; grouped control-valve and dual pushbutton stations; headstock driven by adjustable-speed motor; telescoping guide covers; motors and controls built-in. Landis Tool Co., Waynesboro, Pa.

GRINDER. Convertible to internal or external type. Hydraulically operated worktable controlled by micro-switch operated solenoid valves. Cross slide hand operated. Arter Grinding Machine Co., Worcester 5, Mass.

SUMP CLEANER. Removes chips, grindings, cutting oil, etc., from machine tool sumps. Also dispenses clean liquids. Automatic cutout of motor and pump when tank is filled. Unit mounted on large wheels; balanced tank. Tank, frame and wheels are steel; heavy neoprene gasket seals tank lid. Honan-Crane Corp., Lebanon, Ind.

HONING MACHINE. Automatic, multiple-spindle, unit constructed, quill type. Tool expansion and reciprocation of quills powered and controlled hydraulically; positive feed-

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This advertisement is not an announcement of "bargain" prices. The Alsimag price structure remains unchanged. It is simply a statement of fact and an invitation to submit your insulator problems to Alsimag for cost and design analysis. You may be surprised to find Alsimag production efficiency enables you to buy superior insulators at a price competitive with materials which you have always thought of as "cheap."

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